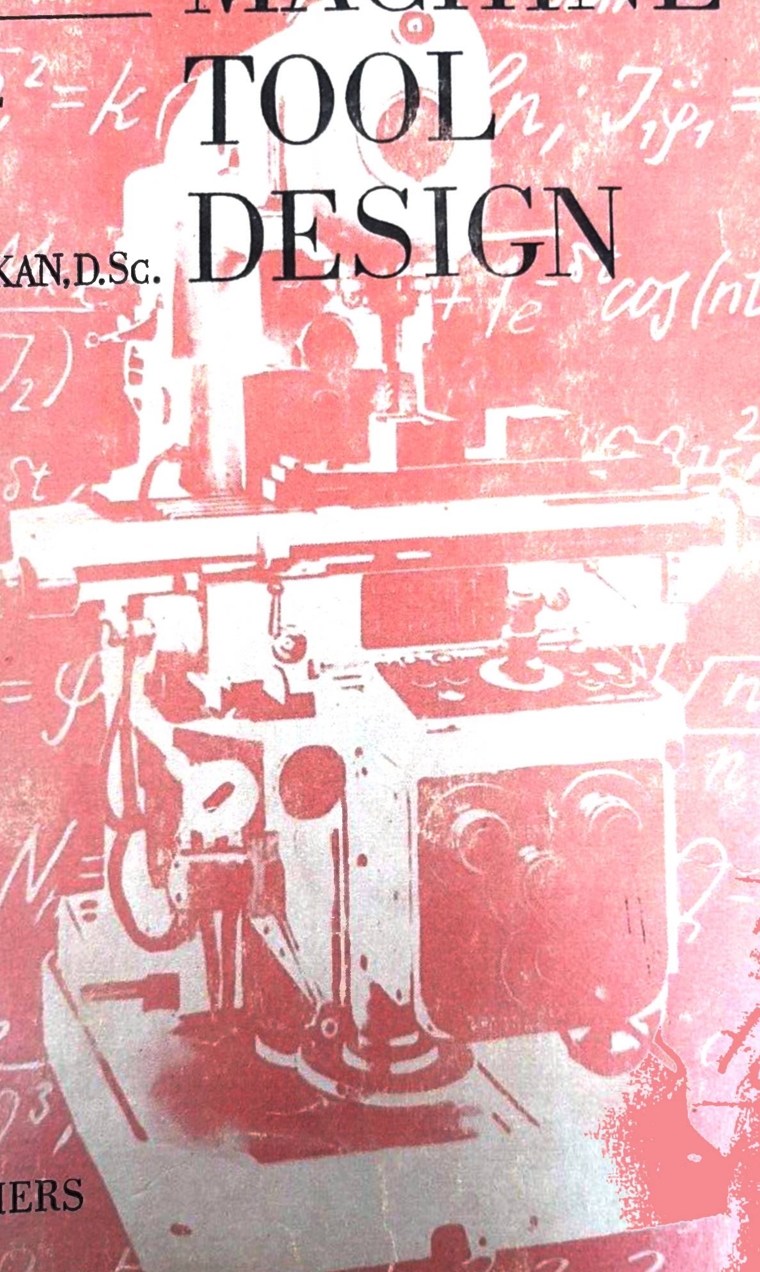
VOL. 3

MACHINE TOOL DESIGN

GENERAL EDITOR

N. ACHERKAN, D.Sc.



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Volume 3 of the fundamental work MACHINE TOOL DESIGN contains Part Five. The complete work is to be published in four volumes.

Part Five deals with machine tool design proper. It is a comprehensive scientific treatment of the subject and is a revised and complemented version of a previous Russian edition which has become a reliable reference book for all Soviet machine tool engineers and has been translated into French. Such questions as performance criteria, basic design data, principal specifications and the development of the kinematic scheme of a new machine tool are dealt with in great detail.

Design recommendations are given as well as the necessary calculation data for the basic elements of machine tools—speed and feed gearboxes, stepless drives, rapid traverse mechanisms, spindles and spindle bearings, mechanisms for rectilinear motion, small displacement and periodic motion, reversing devices, beds, columns, tables and other housing-type components, slideways and antifriction ways.

Of great interest are the chapters on machine tool control systems and dynamic calculations and analysis in machine tool design.

Part Five was written by Professor N. Acherkan, D. Sc. (Eng.), Professor V. Push, D. Sc. (Eng.), N. Ignatyev, Candidate of Eng. Sc. and V. Kudinov, Candidate of Eng. Sc., all from the faculty of the Machine Tool Engineering Institute in Moscow.

This fundamental four-volume work was written by scientists and specialists on the teaching staff of the Machine Tool Engineering Institute in Moscow.

The editor, Prof. N. Acherkan D. Sc. (Eng.), is an eminent Soviet specialist in machine tool design and the author of over 50 scientific works and textbooks in this field. Prof. Acherkan holds the title of Honoured Scientist of the Russian Federation and has occupied the chair of Machine Tool Design for 35 years.

Prof. Acherkan graduated in 1914 from the Warsaw University where he majored in mathematics. He received a gold medal for work done in the university. In 1920 he received a second degree from the mechanical engineering department of the Petrograd Polytechnical Institute.

The present work was translated from the considerably revised and recently published second edition which includes all the latest achievements in world machine tool practice. The earlier edition has become an indispensable handbook for Soviet designers. Therefore, the publishers feel that this work should be of great value to engineers engaged in the design, manufacture and maintenance of machine tool equipment. It can also be used to advantage by the students of engineering institutes majoring in Process Engineering, Metal-Cutting Machine Tools or Cutting Tool Design.

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ИЗДАТЕЛЬСТВО «МАШИНОСТРОЕНИЕ» МОСКВА

MACHINE TOOL DESIGN

N. ACHERKAN, V. PUSH, N. IGNATYEV, V. KUDINOV

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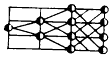

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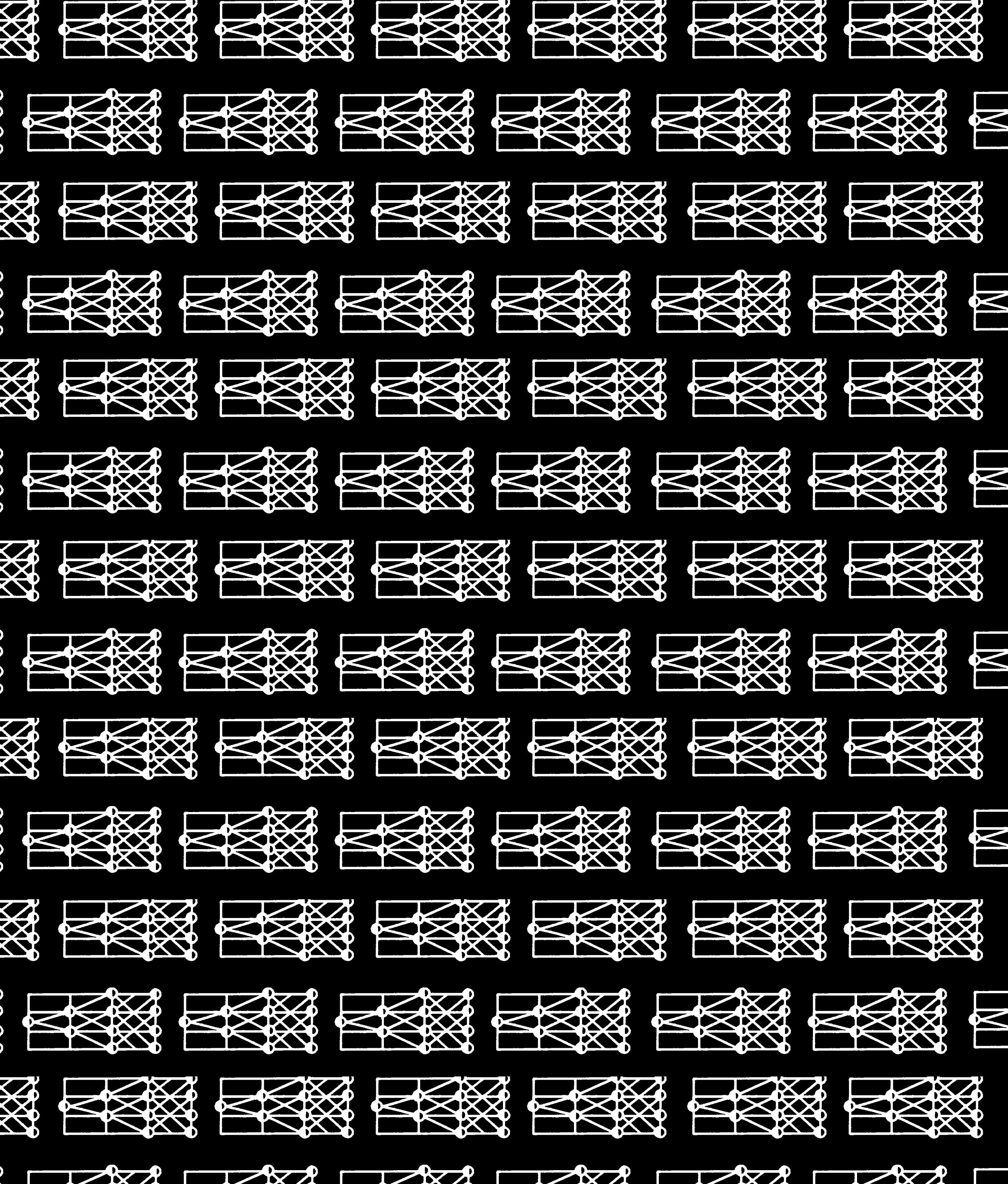


CHAPTER 1

PERFORMANCE CRITERIA. BASIC DATA FOR MACHINE TOOL DESIGN

1-1. Machine Tool Performance Criteria

Ever advancing industry makes higher requirements, year by year, to machine tool performance. The principal performance criteria that must be taken into consideration in designing a new machine tool are: safety and ease of operation, accuracy, dependability, production capacity, amount of material required in manufacture, producibility, production costs, and the level of operating expenses. Not all of these can be expressed in the form of quantitative indices at the present time.

Primarily, the machine tool designing engineer must ensure entirely safe operation, provide maximum ease and convenience of controls, and incorporate features enabling the blank to be readily loaded and clamped, and the finished work to be easily removed.

Reliable protection for the operator, not only against *accidents*, but against *excessive fatigue* as well, is a must in modern machine tools. No pilot model can be put into lot production unless this requirement is complied with.

The higher the level, or degree, of *automaticity* in a machine tool, the simpler and easier it is to operate. The raising of this level up to complete automation of the whole production cycle, including in-process gauging of the workpiece dimensions, feedback tool resetting, loading of blanks and unloading the finished work or semifabricated parts, is one of the most prominent trends in modern machine tool engineering (see Part Six, Vol. 4).

The *operating accuracy* of a machine tool must be such that work of the specified accuracy in size and shape, i.e., within the required dimensional and geometrical tolerances, can be efficiently produced during its whole service life. The operating accuracy is determined by the geometrical, kinematic and dynamic accuracy (see Parts Five and Seven) or, in other words, the capacity of retaining its shape and dimensions constant with adequate stability under the action of the maximum cutting forces, workpiece weight and the counteracting forces and torques developed by the first two factors. The required operating accuracy is achieved by a proper design layout, ample rigidity of the units and of the machine tool as a whole, and the vibration-proof features incorporated in the design.

The *dependability*, or *serviceability*, of a machine tool, as that of any other system, can be defined as its capacity to do its specified job, and is determined

mainly by its failure-proof qualities and the convenience and economical features of its maintenance. Quantitatively, serviceability should be expressed by probability characteristics and parameters. As yet, however, there is no practically suitable method for calculating this index, neither for a separate machine tool, nor for an automatic or semiautomatic production line or transfer machine. The serviceability depends, for all practical purposes, both on the construction of a machine tool, and in particular on the complexity of its kinematic structure and on its producibility, and to an even greater extent on the quality of its manufacture and assembly.

The *production capacity* of a machine tool is characterized by its ability to machine a definite number of workpieces, or a definite area of workpiece surface, in a unit of time in accordance with the given specifications.

The *absolute production capacity* (after Academician V. Dikushin) is defined as the amount of usefully expended power of the machine tool per workman required for its normal operation, and can be expressed by the equation

$$N_T = N_c + N_h = \frac{1}{6120}\left(\sum_{i=1}^{n} P_c v_c \frac{T_{ci}}{T_{cy}} + \sum_{i=1}^{k} P_h v_h \frac{T_{hi}}{T_{cy}}\right) \text{ kW} \tag{1}$$

where N_c and N_h = power, kW, expended on the cutting action and on handling operations, respectively

P_c and v_c = force, kgf, and speed, m per min, respectively, associated with each working motion of the machine tool

P_h and v_h = force, kgf, and speed, m per min, respectively, associated with each handling motion

T_{ci} and T_{hi} = duration, min, of the corresponding working and handling motions, respectively

T_{cy} = total cycle time, min.

Present-day machine tools are distinguished for their very high absolute production capacity (up to 100 kW and even more).

This index requires involved calculations since it is necessary first to determine all the useful forces acting in the machine tool. Hence, other, approximate indices are commonly employed for a comparative assessment of machine tool production capacity.

The *cutting capacity* is defined as the amount of metal or other material (in weight or volume units) removed from the blank in unit time. This productivity index can be used as an approximate comparative assessment of general-purpose machine tools designed for roughing operations with a large $\frac{T_c}{T_{cy}}$ ratio, where T_c and T_{cy} are the same as in equation (1).

The *formative capacity* is defined as the area of the surface machined in unit time. This index is convenient for comparing the productivity of general-purpose machine tools designed for finishing operations.

The *piece output* (theoretical) denotes the number of workpieces machined in unit time and is the most convenient index for assessing the production capacity of single-purpose and special machine tools. The number of workpieces produced in unit time is calculated by the formula

$$Q=\frac{1}{T_{cy}}=\frac{1}{T_c+T_h}\text{ pcs per min} \tag{2}$$

The *material requirement* of a machine tool (or *metal requirement*, which is almost the same since the share of nonmetallic materials is still very small in the weight of a machine tool) is evaluated in the USSR as the amount of metal (by weight) in the machine tool per unit of its power employed in the working process. Thus

$$M=\frac{G}{N_c}\text{ kgf per kW} \tag{3}$$

where G = weight of the machine tool, kgf
N_c = power of the main drive, kW.

The material requirement of modern general-purpose machine tools ranges from 200 to 1000 kg per kW. This quality-of-design index is used to compare machine tools of the same type.

As more and more refined constructions are being developed and more exact calculation procedures are devised (enabling the actual safety factors, rigidity margin, etc., to be reduced) the material requirement of machine tools should decrease.

The *producibility* of a machine tool (or any other machine or structure) is characterized by the degree of complexity in the manufacture and assembly of its units, components and the whole machine tool. A rough approximation of the producibility can be made by considering the number of unique parts, designed for the particular model, and the number of parts covered by government standards (GOST in the USSR), engineering industry standards or, if the manufacturer is known, plant standards. Producibility of a machine tool depends upon many factors, including the scale of production (lot size) and the process engineering level of the manufacturing plant. Moreover, producibility may vary in the course of time due to the introduction of new, advanced manufacturing techniques.

One of the most essential objectives of the designer is to ensure *minimum production cost* of the machine tool being developed, under the condition that all the specifications have been complied with. This is achieved by proper design layout; choice of the optimum construction between the possible versions of each unit; selection of materials possessing the necessary and adequate physicomechanical properties, without the misuse, for example, of high-quality steels, nonferrous metals, etc.; and by assigning the necessary and adequate dimensions to the parts on the basis of calculations made with the highest possible accuracy and with judiciously limited safety margins.

The production cost also depends on the production management system employed in the manufacturing plant.

The *appearance* of machine tools is receiving much attention in recent years. If the machine tool being developed has an advantageous design layout, i.e., if its units and component assemblies are properly positioned in relation to each other, it will have a sufficiently good shape from the point of view of industrial design principles. The appearance can be further improved by redesigning the intersections of the external surfaces, the surface of the covers, etc. These measures, however, should have no adverse effects on the operational features of the machine tools, in particular on the convenience of servicing and control, or on its producibility. The colour or colours of machines should also comply with aesthetic principles if the colours conform to psychophysiologic requirements as well. In any case, the measures taken to satisfy the principles of industrial design should not increase the production cost of the machine tool, or expenditures for its operation and maintenance.

Of all the features of the machine tool being designed, the critical ones, in conjunction with complete safety of operation, are high production capacity, and adequate accuracy and surface finish of the machined work.

In the Soviet Union, each new model must conform to the approved type and size range of metal-cutting machine tools, while its capacity and other specifications must be in accordance with the pertinent USSR standards. Standards stipulating the principal capacities and dimensions, accuracy standards and rigidity standards are available for each type of machine tool according to its class of accuracy (see Part Seven, Vol. 4).

In addition to the requirements mentioned above, common to all machine tools, the engineering assignment for the design of a new model may include special requirements which influence the design layout and construction of the machine tool and consequently must be taken into consideration by the designing engineer. Such additional requirements are most often made in reference to special-purpose and specialized machine tools.

If the material of the parts has been correctly chosen, with due regard for the operating conditions, and design calculations have been done with sufficient accuracy, such factors as *wear resistance* of the elements that primarily affect performance (ways, spindle journals in sleeve bearings, sleeve and antifriction spindle bearings, lead screws and nuts, etc.) and a suitably long *service life* will be ensured.

1-2. Trends in the Development of Modern Machine Tools

The general course in the development of modern machine tool engineering can be characterized by the quest for the highest possible productive capacity under the condition that the necessary and adequate machining accuracy

is ensured and, for finishing operations, the necessary and adequate surface finish is obtained as well on the machined workpieces. The consequent tendencies are toward: (a) a reduction in piece time, (b) the prevention of deformation in the machine-tool-workpiece complex leading, in operation, to deviations in the dimensions and geometrical shape of the workpiece that are not within the specified tolerances, and (c) the prevention of vibrations in the machine-tool-workpiece complex resulting in undue surface roughness of the parts being machined in the machine tool.

A number of the most essential special trends, typical of modern machine tool engineering, follow from these general tendencies:

1. The speeds of the cutting and feed motions are being increased to reduce the machining time. This has led to an increase in the relative number of models for high-velocity machining.

2. The power of the main drives is being increased to accommodate the higher cutting speeds and, frequently, the heavier chip being removed and the greater number of simultaneously operating cutting tools.

3. More extensive systems of infinitely variable (stepless) speeds and feeds are being applied, enabling the optimum cutting conditions to be set up and changed as required without stopping the machine. The purpose of such systems is to reduce the machining time.

4. Machine tools are being equipped with a great variety of auxiliary devices whose aim is to reduce the part of the handling time that is not overlapped by the machining time. Included here are devices for quickly changing cutting tools, mechanized fixtures for rapidly changing the position of the work, automatic in-process gauging devices with feedback tool adjustment, devices for facilitating blank loading and removal of the finished workpiece, etc.

Many machine tools are furnished with devices which extend their range of application and thereby, frequently, exclude the need for machining the work consecutively in several machine tools.

5. The working cycle is being automated to set up a positive working pace, constant for the given setup, for the purpose of reducing the handling time and to ensure the required machining accuracy irrespective of the skill of the operator. In connection with this trend, various methods and means of automation have found wide application. This includes programme-control systems and particularly numerically controlled machine tools. The incorporation of such systems greatly affects the design of machine tools (see Part Six, Chapters 9 through 15, Vol. 4).

6. Machine tool controls are being simplified, mainly by automation of the working cycle and the application of various interlocking devices (see Chapter 11 and Part Six).

7. The static and dynamic rigidity of machine tools and their vibration-proof properties are being increased as measures required to raise cutting

speeds and feeds and the power of the drive, complying, at the same time, with the high requirements as to shape and dimensional accuracy, and surface finish of the machined work.

8. There is a wide, ever-increasing application of electrotechnics, electronics, hydraulics and pneumatics for performing various functions. It may be assumed that fluidics will find application in machine tool controls due to its compactness, fast response, simple manufacture and, therefore, low cost in lot production.

9. Standardized parts and units are being used to the maximum extent to reduce the designing and lead time in the production of a new model, as well as the production costs.

This trend is evident in its most fundamental and practical form in the production of unit-built machine tools which are designed on the basis of standard parts and units with the addition, in each separate case, of certain special devices. This trend can also be observed in the change-over to unitized design in which the machine tool is developed as a combination of self-contained special-purpose units, employed for performing the same definite function in different types of machine tools. An application of the principles of unitized design permits a variety of special-purpose and specialized machine tools to be developed from a single basic design. The engineering of a new model, in this case, is much simpler and requires less lead time due to the unification of the units making up the basic and modified machine tools.

10. An increase is being observed in the relative amount of machine tools designed for efficient multiple-tool machining as, for example, multiple-tool lathes, multiple-spindle drilling, milling and boring machines (unit-built types), etc.

11. Efforts are being made to utilize machine tools more fully, especially expensive ones tended by high-skilled operators and occupying considerable floor space in the shop. This particularly refers to heavy machine tools. As a result, measures are being taken to increase the adaptability of such machine tools to various jobs.

Examples are the planer-type milling machines which can operate as well at planing speeds and feeds; planers on which milling heads can be mounted; and large vertical turning and boring mills with work tables which can reciprocate, in addition to rotation in one direction, enabling machining time to be sharply reduced in turning large segments of complex configuration. These turning mills can also be used as rotary dividing machines.

12. Machine tools for high-velocity machining, which remove a large amount of metal from the blank, are being furnished with devices for automatically disposing of the hot chips. The latter are not only a hazard to the

operator and make the machine more difficult to handle, but may damage the ways and other critical parts.

Certain trends are observed only in specific groups of machine tools. Thus, optical devices are being more extensively used in precision and high-accuracy machine tools for setting up the relative positions of the tool and work to a greater degree of accuracy, and for reading off co-ordinate dimensions (jig boring machines, profile grinders and certain milling machines). Automatic sizing devices are finding widespread application in grinders. A tendency to increase metal removal is observed in the group of microfinishing (honing and lapping) machines.

It follows from the above that the requirements of mass production techniques have a profound effect on the development of machine tool design. The same factor is responsible for the large increase in the share of special and specialized machine tools among the metal-cutting equipment of up-to-date engineering plants.

One of the most distinctive features of modern machine tool engineering is the development of automatic groups of machine tools, transfer machines and whole automatic shops and even plants for the line production of machine components (see Part Six, Chapters 16 through 21).

1-3. Machine Tool Design Recommendations

Along with the production-economics indices, machine tool design is based on the general working capacity criteria of the principal units and parts. Such criteria include static and fatigue strength, wear resistance, rigidity, vibration-proof properties, and temperature conditions.

In designing certain units of a machine tool it is necessary tentatively to determine the magnitude, direction and kind of forces acting during various periods of operation and, in particular, during the periods of transient motion (starting, braking and reversing). This enables a design schedule to be drawn up for making the necessary calculations to design the mechanisms and units of the machine tool.

In drawing up such a schedule, the following acting loads should be taken into consideration:

1. *Motive power of the drive*. In calculations, these forces are taken in accordance with the rated power or available torque of the drive motor. Instructions for selecting the power of the drive motor are given below (Secs. 2-5). The motive power of the drive is dependent on the characteristics of the electrical, hydraulic or pneumatic drive employed in the given machine tool.

2. The *cutting forces* are represented in the form of three components P_z, P_x and P_y*; methods of determining them are given in the study course "The Cutting of Metals"**.

In the great majority of cases, all the acting forces are variable (with the exception of the force due to the weight of blanks that are not unduly heavy and have a small machining allowance), and the range of their variation can reach considerable values. This circumstance is taken into consideration by introducing dynamic coefficients in the equations for the corresponding design forces and torques, i.e.,

$$P_{des} = k_{P\,dyn} P_{st} \quad \text{and} \quad M_{des} = k_{M\,dyn} M_{st} \tag{4}$$

in which

$$k_{P\,dyn} = 1 + \frac{P_{dyn}}{P_{st}} \quad \text{and} \quad k_{M\,dyn} = 1 + \frac{M_{dyn}}{M_{st}} \tag{5}$$

where P_{st} and P_{dyn} = static and dynamic forces, respectively
M_{st} and M_{dyn} = static and dynamic torques, respectively.

For this reason, all critical parts of machine tools are designed on the basis of their fatigue strength.

If the cutting forces are not known and may vary in fairly wide ranges (as in designing general-purpose machine tools), they can be determined approximately by the formulas

$$\left.\begin{aligned} P_z &= k\,(a + 0.4c)\,b \text{ kgf} \\ P_N &= \sqrt{P_x^2 + P_y^2} = k\,(0.4a + c)\,b \text{ kgf} \end{aligned}\right\} \tag{6}$$

where P_z and P_N = components of the cutting force, kgf
k = characteristic of the material to be machined, kgf per sq mm (for steel $k \cong 120$ to 180, depending upon its hardness; for cast iron $k \cong 90$ to 110)
a and b = thickness and width, respectively, of the undeformed chip, mm
c = width of the band of contact on the tool flank, which can be taken for calculations as one half of the permissible wear band on the flank, mm.

The ratio of the components P_x and P_y depends upon the form of the cutting edge.

*In standard metal cutting notation (USSR), P_x, P_y and P_z are the axial (longitudinal), radial (normal) and vertical (tangential) components of the cutting force, respectively, as referred to a lathe tool.

**The magnitude and nature of variations in the cutting force components with time depend upon the chip-forming process and are considered in detail in the same study course.

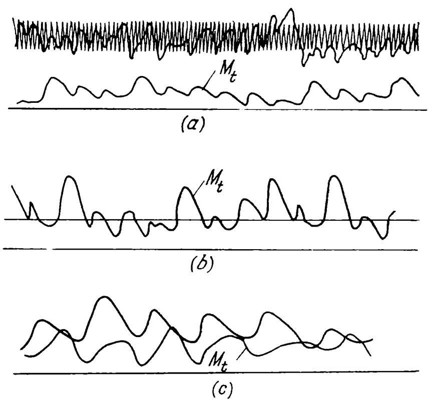


Fig. 1. Oscillograms of the torque developed in gear hobbing

The cutting forces are of a markedly variable nature in some types of machining, such as milling and gear hobbing. For example, the cutting forces vary by 50 per cent or even more (Fig. 1) in gear hobbing. The variability of the forces involved in the cutting process is usually characterized by the ratio of the amplitude of force variation to the average value.

3. *Friction forces* are usually taken proportional in machine tool design to the normal load on the friction surfaces. The appropriate coefficients of friction depend upon many factors, and primarily upon the material and condition of the friction surfaces, lubrication conditions and sliding velocity. In the absence of a lubricant (dry friction), the coefficient of friction $f = 0.2$ to 0.3 (and sometimes more); for semifluid friction $f \cong 0.03$ to 0.2, and for fluid friction $f \leqslant 0.002$ to 0.05. Rolling friction is frequently characterized by the ratio of the coefficient of rolling friction to the radius of the rolling member. This ratio is very small for hardened steel.

4. *Inertia loads* are to be taken into consideration for all transient processes (starting, braking, etc.). The following formula can be used to calculate the moment of inertia referred to the motor shaft (if friction losses are neglected)

$$J_{ref} = \sum_k J_k \left(\frac{\omega_k}{\omega_1}\right)^2 + \sum_i m_i \left(\frac{v_i}{\omega_1}\right)^2 \tag{7}$$

where J_k and ω_k = inertia moment and angular velocity, respectively, of the rotating masses

m_i and v_i = mass and linear velocity, respectively, of reciprocating masses

ω_1 = angular velocity of the motor shaft.

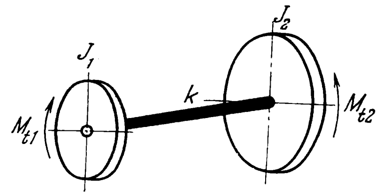


Fig. 2. Diagram of a two-mass system with an elastic connection

It usually proves sufficient to take the mass of the motor and operative member of the machine tool (spindle, table, faceplate, etc.) into account, neglecting the masses of the intermediate transmitting mechanisms.

5. *Reactions at the supporting surfaces* are determined from equations of equilibrium. If necessary, in solving statically indeterminate problems (referring to multiple-support spindles and shafts, straight ways and guides, etc.), supplementary deflection equations are worked out.

It is permissible in the majority of cases, with sufficient accuracy for all practical purposes, to regard reactions as concentrated forces on the basis of the linear law of pressure distribution. If support surfaces are of comparatively small extent, it is assumed that the pressure is uniformly distributed, i.e., that the resultant force is applied at the middle.

6. *Forces due to starting and braking.* In calculating starting and braking torques, the kinematic chain of the machine tool drive can be reduced, in most cases, to a design diagram consisting of two masses linked together by an elastic connection (Fig. 2). Then, if an external starting (or braking) torque M_{t1} is applied to the mass with the inertia moment J_1, and an additional torque M_{t2} due to the friction forces is applied to the mass with the inertia moment J_2, the motion equations in starting (or braking) can be written as

$$\left.\begin{aligned} J_1\ddot{\varphi}_1 + c\,(\dot{\varphi}_1 - \dot{\varphi}_2) + k\,(\varphi_1 - \varphi_2) &= M_{t1} \\ J_2\ddot{\varphi}_2 - c\,(\dot{\varphi}_1 - \dot{\varphi}_2) - k\,(\varphi_1 - \varphi_2) &= \pm M_{t2} \end{aligned}\right\} \qquad (8)$$

where φ_1 and φ_2 = angles of rotation of the masses with inertia moments J_1 and J_2, respectively
c = damping coefficient
k = rigidity of the shaft.

The lower sign preceding M_{t2} refers to starting; the upper sign refers to braking.

After dividing the first of these equations by J_1 and the second by J_2, and after subtracting the second equation from the first, we obtain

$$\ddot{\psi}+c\,\frac{J_1+J_2}{J_1J_2}\,\dot{\psi}+k\,\frac{J_1+J_2}{J_1J_2}\,\psi=\frac{M_{t1}J_2\mp M_{t2}J_1}{J_1J_2} \tag{9}$$

where $\psi=\varphi_1-\varphi_2=$ angle of twist of the shaft.

The solution of this equation for the case in which $M_{t1}=$ const and $M_{t2}=$ const is

$$\psi=\frac{M_{t1}J_2\pm M_{t2}J_1}{k\,(J_1+J_2)}+e^{-\delta t}\,(C_1\sin nt+C_2\cos nt) \tag{10}$$

where $n=\sqrt{\dfrac{k\,(J_1+J_2)}{J_1J_2}}=$ natural frequency of vibrations of a two-mass elastic system according to Fig. 2

$\delta=$ value characterizing damping.

Making the following substitution

$$\frac{M_{t1}J_2\pm M_{t2}J_1}{k\,(J_1+J_2)}=A \tag{11}$$

then

$$\psi=A+e^{-\delta t}\,(C_1\sin nt+C_2\cos nt) \tag{12}$$

At the initial conditions $\psi_{t=0}=0$ and $\dot{\psi}_{t=0}=0$, we can write

$$C_2=-A \quad \text{and} \quad C_1=C_2\frac{\delta}{n}=-A\frac{\delta}{n} \tag{13}$$

and equation (12) becomes

$$\psi=A\left[1-\frac{e^{-\delta t}}{n}\,(\delta\sin nt+n\cos nt)\right] \tag{14}$$

Upon introducing the values B and β, defined by the conditions

$$\delta=-B\sin\beta \quad \text{and} \quad n=B\cos\beta$$

and, consequently,

$$B=\sqrt{\delta^2+n^2} \quad \text{and} \quad \tan\beta=-\frac{\delta}{n} \tag{15}$$

we obtain

$$\psi=A\left[1-\frac{e^{-\delta t}}{n}\,B\,(-\sin nt\sin\beta+\cos nt\cos\beta)\right]=$$

$$=A\left[1-\frac{\sqrt{\delta^2+n^2}}{n}\,e^{-\delta t}\cos(nt+\beta)\right] \tag{16}$$

or, in the final form,

$$\psi=\frac{M_{t1}J_2\pm M_{t2}J_1}{k\,(J_1+J_2)}\left[1-\sqrt{\left(\frac{\delta}{n}\right)^2+1}\;e^{-\delta t}\cos(nt+\beta)\right] \tag{17}$$

where $\beta=\arctan\left(-\dfrac{\delta}{n}\right)$.

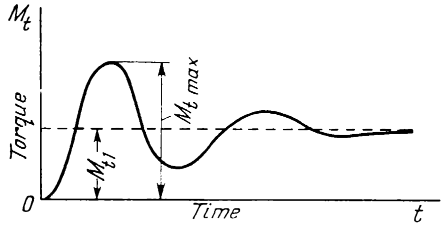


Fig. 3. Curve of the torque according to equation (19)

If the ratio of the damping characteristic to the natural frequency of vibration $\frac{\delta}{n} \to 0$, then $\sqrt{\left(\frac{\delta}{n}\right)^2 + 1} \to 1$, $\beta \to 0$ and then

$$\psi = \frac{M_{t1}J_2 \pm M_{t2}J_1}{k(J_1 + J_2)}(1 - e^{-\delta t}\cos nt) \tag{18}$$

This equation enables the true value of the torque, transmitted through the kinematic chain of the drive during starting or braking, to be determined. Thus

$$M_t = k\psi = \frac{M_{t1}J_2 \pm M_{t2}J_1}{J_1 + J_2}(1 - e^{-\delta t}\cos nt) \tag{19}$$

The last equation is presented graphically in Fig. 3.

Equation (19) can be used to determine the maximum torque developed in the kinematic chain of the drive in starting. Indeed, if damping is disregarded ($\delta = 0$), then, for starting, the last equation will be

$$M_{t\,max} = 2\frac{M_{t1}J_2 + M_{t2}J_1}{J_1 + J_2} \tag{20}$$

If, for example, $J_2 \gg J_1$ (true for most of the heavy machine tools), then

$$M_{t\,max} \cong 2M_{t1} \tag{21}$$

Thus, the maximum torque developed in the elastic system of the drive during the starting period is twice the motor torque.

The general motion equations of the drive given above are also applicable in designing various units of the machine tool subject to transient processes.

In calculations for strength, rigidity and wear resistance, a properly drawn-up design diagram, in which all the acting forces have been taken into consideration, enables the stresses, deflections and pressure on the friction surfaces to be determined, after which they can be compared with the accepted standard values.

The rigidity of movable and fixed joints in the various parts and units of the machine must be taken into account in determining deformation. In this case, the static rigidity of the joint (ratio of the load to the consequent deformation) and the general rigidity of the elastic system are usually assumed to be constant. This assumption is justified in practice by calculations and experimental data (Fig. 4) showing the approximate linear relationship between forces and deformations in complex elastic systems.

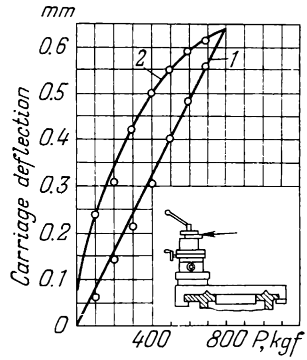


Fig. 4. Elastic deflection of a lathe carriage vs load: 1—loading curve; 2—unloading curve

In comparing the values found by calculation with the permissible values, a factor of safety, or design factor, is introduced. This factor takes into account the degree of accuracy of the calculations.

Temperature deformations are determined for the working temperature which is found by solving the heat balance equation.

A great many elements of machine tools cannot be calculated with sufficient accuracy by existing formulas. This is due, on the one hand, to the extremely complex shape of many machine parts and, on the other hand, to the complex system of acting forces and torques, and their variation in value and direction during operation. Therefore, to determine sufficiently dependable and, at the same time, economical dimensions of such parts (this being of especial importance in special-purpose and heavy machine tools), recourse is had, more and more frequently in recent years, to experimental investigations on scale models of the parts in question. The results of the experiments are then transferred to the part, using the formulas of the theory of mechanical similitude which are based on the well-known relationship between the similarity scales

$$\lambda\mu\tau^{-2}\varphi^{-1} = 1 \tag{22}$$

where λ = scale of linear dimensions
μ = scale of masses
τ = time scale
φ = scale of forces.

The conversion factor for the required characteristic or parameter, as well as the relationship between the scales for the conditions of the experiment are determined on the basis of the corresponding equations and dimensionality of this characteristic. For example, the initial equation for determining

the bending strain of a part, that can be regarded approximately as a beam, is

$$\frac{d^2y}{dx^2} = -\frac{M}{EI}$$

where $y = f(x)$ = equation of the bent axis of the part (beam) in x-y co-ordinates
M = bending moment
E = modulus of elasticity of the material of which the part is to be made
I = moment of inertia of the beam cross-sectional area.

Hence, for the model (subindex m)

$$\frac{d^2y_m}{dx_m^2} = -\frac{M_m}{E_m I_m} \tag{23a}$$

and for the actual part (subindex a)

$$\frac{d^2y_a}{dx_a^2} = -\frac{M_a}{E_a I_a} \tag{23b}$$

However, since

$$y_a = \lambda y_m; \quad x_a = \lambda x_m; \quad M_a = \lambda\varphi M_m \quad \text{and} \quad I_a = \lambda^4 I_m$$

equation (23b) can be written as

$$\frac{1}{\lambda}\frac{d^2y_m}{dx_m^2} = -\frac{\varphi M_m}{E_a\lambda^3 I_m} \quad \text{or} \quad \frac{d^2y_m}{dx_m^2} = -\frac{\varphi}{\lambda^2}\frac{M_m}{E_a I_m} \tag{23c}$$

and it follows from equations (23a) and (23c) that the scale of forces φ and scale of linear dimensions λ are related in the given case by the equation

$$\varphi = \lambda^2 \frac{E_a}{E_m} \tag{24}$$

If the model and the actual part are of the same material, then the load scale $\varphi = \lambda^2$. A selection of these two scales, independently of each other, restricts the selection of the material for making the model by the condition that $E_m = \frac{\lambda^2}{\varphi} E_a$.

If these conditions of the experiment are complied with, the deflection of the machine part will be $y_a = \lambda y_m$, where y_m is the experimentally established deflection of the model in the same (homological) cross section.

In a similar way, using the law of dynamic similarity for parts which can be treated as round shafts, the natural frequency of torsional vibrations can be determined proceeding from the equation

$$\frac{\partial^2\theta}{\partial x^2} = \frac{\rho}{G}\frac{\partial^2\theta}{\partial t^2} \tag{25}$$

where θ = angle of twist
ρ = density of the material
G = shear modulus of elasticity
t = time.

Hence, for the model and actual part, respectively:

$$\frac{\partial^2\theta_m}{\partial x_m^2} = \frac{\rho_m}{G_m}\frac{\partial^2\theta_m}{\partial t_m^2} \quad \text{and} \quad \frac{\partial^2\theta_a}{\partial x_a^2} = \frac{\rho_a}{G_a}\frac{\partial^2\theta_a}{\partial t_a^2} \qquad \text{(25a and b)}$$

Taking the scales into account, equation (25b) can be written as

$$\frac{1}{\lambda^2}\frac{\partial^2\theta_m}{\partial x_m^2} = \frac{\rho_a}{G_a}\frac{1}{\tau^2}\frac{\partial^2\theta_m}{\partial t_m^2} \qquad \text{(25c)}$$

while the required relationship between the scales of linear dimensions λ and time τ follows from equations (25a) and (25c). Thus

$$\tau = \lambda\sqrt{\frac{G_m}{G_a}\frac{\rho_a}{\rho_m}} \qquad (26)$$

To determine the flexural rigidity, expressed by the ratio of the force to the deflection (dimensionality of kgf per micron), the conversion factor is $\frac{\varphi}{\lambda}$; for the frequency of vibrations (dimensionality of $\sec^{-1}$), the conversion factor is τ^{-1}, etc.

Numerous experiments have proved that the method of mechanical simulation gives sufficiently accurate results for practical purposes, for example in predicting deformation of a designed machine part, namely its flexural and torsional rigidity, main frequency of its natural vibrations, type of vibrations, etc.

Engineering design procedures and methods, based on theoretical analysis and experimental research, have been worked out by ENIMS* for the most complex and critical parts and units of machine tools, including beds and columns, slideways, roller ways, hydrostatic ways, spindle units with sleeve, antifriction, hydrostatic and aerostatic bearings, lead screws with ordinary and ball-bearing nuts, etc.

*Experimental Research Institute for Metal-Cutting Machine Tools (Moscow).

CHAPTER 2

DETERMINING THE PRINCIPAL SPECIFICATIONS OF THE MACHINE TOOL BEING DESIGNED

2-1. Selecting the Maximum and Minimum Cutting Speeds and Feeds

Maximum and minimum cutting speeds and feeds, for machining blanks of the maximum and minimum sizes that are to be accommodated, are selected by analyzing the manufacturing process. In designing general-purpose machine tools, the initial data can be found in the manufacturing process for typical workpieces that are to be machined in the given machine tool.

The maximum and minimum cutting speeds and feeds should be established for all operations with all the different types of cutting tools that are to be used. The aim of this process analysis is not only to determine the spindle speed and the feed ranges (kinematic features), but also to find which operations and machining conditions require the most power from the drive, the highest spindle torques and the maximum feed forces (power features).

In designing machine tools, the maximum and minimum values of the specifications are determined on the basis of the maximum and minimum values of only a few machining conditions, but ones taken from different operations. For instance, in designing a turret lathe, the maximum power of the drive is determined for rough turning with carbide-tipped tools; the maximum spindle torque for rough turning with high-speed steel tools; and the maximum feed force on the turret slide or saddle for drilling at maximum capacity.

Thus, the maximum and minimum cutting speeds and feeds for various operations are determined by tying them in with definite specifications of the machine tool. At the same time, it is necessary to take into consideration possible future advances in machining techniques and improvements in cutting tool design, making provision for them in the specifications of the new machine tool.

In assessing the maximum and minimum cutting speeds and feeds and the corresponding specifications obtained as a result of such analysis, it is also necessary to give due consideration to the place allotted to this machine tool in the size range of the given group, and the possibility of machining work of the maximum sizes with maximum cutting speeds and feeds in the adjacent sizes of this machine tool group.

2-2. Series of Spindle Speeds for Machine Tools

The extreme values of spindle speeds n_{max} and n_{min} can be determined for machine tools with a rotary primary cutting motion if the extreme diameters d_{max} and d_{min} to be cut are known, and the maximum and minimum cutting speeds v_{max} and v_{min} have been established for the given diameters. Thus

$$n_{max} = \frac{1000 v_{max}}{\pi d_{min}} \quad \text{and} \quad n_{min} = \frac{1000 v_{min}}{\pi d_{max}} \tag{27}$$

Then the *range ratio of spindle speed variation* is

$$R_n = \frac{n_{max}}{n_{min}} \tag{28}$$

Hence

$$R_n = \frac{n_{max}}{n_{min}} = \frac{v_{max}}{v_{min}} \frac{d_{max}}{d_{min}} = R_v R_d \tag{29}$$

if $\frac{v_{max}}{v_{min}}$ is denoted by R_v and $\frac{d_{max}}{d_{min}}$ by R_d.

It is evident that R_n depends only upon the ratio of the maximum and minimum diameters and cutting speeds involved in machining. Making allowances for possible future improvements in cutting tool design and machining techniques, the value of R_n obtained from equation (29) is increased by approximately 20 to 25 per cent.

To machine work of any diameter d within the indicated limits with the most expedient cutting speed v, it is necessary that the spindle speed $n = \frac{1000v}{\pi d}$, where v is expressed in m per min and d in mm, in all cases. This is only true for *infinitely variable* (*stepless*) *speed variation*, which is achieved by the application of a suitable mechanical, electrical or hydraulic drive of this type (see Sec. 4-6). In the majority of cases, however, modern machine tools are still being designed with a stepped series of spindle speeds.

The problem of the most advantageous distribution of the spindle steps between the extreme values n_{min} and n_{max} was first solved by Academician A. Gadolin (Russian Academy of Sciences) in 1876 on the basis of the following. He proved the advantage of using a *geometrical structure* for the spindle speed series, i.e., one based on a geometrical progression, by showing that the absolute loss of economically expedient cutting speed $(\Delta v)_{max}$ is constant for all intervals in such a speed series. Moreover, the relative loss of cutting speed, $A = \frac{(\Delta v)_{max}}{v}$ for a geometrical series n is also a constant value since for a series (progression) ratio $\varphi = \frac{n_{j+1}}{n_j} = \text{const}$,

$$A = \frac{(\Delta v)_{max}}{v} = \frac{n_{j+1} - n_j}{n_{j+1} + n_j} = \frac{\varphi - 1}{\varphi + 1} = \text{const} \tag{30}$$

If the maximum value of the cutting speed permitted by the whole complex of machining conditions including, in particular, tool life, is chosen as the desirable cutting speed v, then the absolute loss of cutting speed will be

$$\Delta v = v - v_j$$

and the maximum relative loss will be

$$A_{max} = \left(\frac{\Delta v}{v}\right)_{max} = \frac{n_{j+1} - n_j}{n_{j+1}} = \frac{\varphi - 1}{\varphi} \tag{31}$$

or, expressed in per cent,

$$A_{max} = \frac{\varphi - 1}{\varphi}\, 100\% \tag{31a}$$

Thus, the maximum relative loss of cutting speed depends only on φ, the constant ratio of the spindle speed series.

As has been indicated (p. 14), the formative capacity Q is defined as the area of the surface machined in unit time, i.e.,

$$\pi\, dns = 1000 sv \text{ sq mm per min}$$

where s is the feed, mm per revolution (in turning or drilling) or

$$Bns = B\frac{1000}{\pi d} sv \text{ sq mm per min (in milling)}$$

where B is the width of cut, mm.

Therefore, at constant feed, the capacity Q is proportional to the cutting speed. The maximum relative loss of formative capacity for a geometrical series of spindle speeds is

$$\left(\frac{\Delta Q}{Q}\right)_{max} = \left(\frac{\Delta v}{v}\right)_{max} = \frac{\varphi - 1}{\varphi} \tag{32}$$

and is constant.

In the rectangular co-ordinates d and v, a geometrical series is depicted as a diagram with as many rays as there are different speeds, and the following principal relationship exists:

$$n_z = n_1 \varphi^{z-1} \tag{33}$$

where z = number of spindle speed steps

$n_z = n_{max}$

$n_1 = n_{min}$.

Hence

$$\left.\begin{aligned} R_n = \frac{n_z}{n_1} = \varphi^{z-1}; \quad \varphi = \sqrt[z-1]{\frac{n_z}{n_1}} = \sqrt[z-1]{R_n} \\ \text{and} \quad z = 1 + \frac{\log R_n}{\log \varphi} = \frac{\log (R_n \varphi)}{\log \varphi} \end{aligned}\right\} \tag{34}$$

If z is calculated from the last formula, the obtained value is rounded off to a whole number, after which the range ratio of variation R_n is correspondingly changed.

In addition to its economical advantages, geometrical series of spindle speeds have other advantages that are of great importance in designing the machine tool drive (see Chap. 3). For these reasons, geometrical series have found wide application in machine tools.

2-3. Standard Values of the Ratio φ. Standard Series of Spindle Speeds

Standard ratios φ have been established for standard series of spindle speeds in machine tools on the basis of the following:

1. Two-speed three-phase electric motors are frequently employed in the spindle drives of machine tools. The ratio of their synchronous speeds equals 2, for example 3000/1500 or 1500/750. Therefore, if the series of speeds has a member n_x, there must also be a member $n_y = 2n_x$, in which case $n_y = n_x\varphi^E$, where E is a whole number. It follows that

$$n_x\varphi^E = 2n_x \quad \text{and} \quad \varphi = \sqrt[E]{2} \tag{35}$$

2. In Soviet designing offices, account must be taken of USSR Std GOST 8032-56 "Preferred Numbers and Series of Preferred Numbers", as well as Machine Tool Industry Standard H11-1 which establishes preferred values and gradation of parameters in machine tools.

The series of preferred numbers are in the form of geometrical progressions whose constant ratio must comply with the condition

$$\varphi = \sqrt[E]{10} \tag{36}$$

where E is a whole number.

Thus, standard values of φ must satisfy the conditions

$$\varphi = \sqrt[E_1]{2} = \sqrt[E_2]{10} \tag{37}$$

Therefore, $E_1 = 3E'$ and $E_2 = 10E'$, where E' is any whole number.

Hence, and with the addition of the values $\varphi = \sqrt{2} = 1.41$, $\varphi = \sqrt[1]{2} = 2$ and $\varphi = \sqrt[4]{10} = 1.78$, the series of standard values of ratio φ, listed in Table 1, were obtained.

Standard H11-1 permits derivative series to be drawn up by omitting a part of the values in the standard series.

Actual spindle speeds may differ from the tabular values by not more than $\pm 10\,(\varphi - 1)$ per cent.

TABLE 1

φ	1.06	1.12	1.26	1.41	1.58	1.78	2
$\sqrt[E]{2}$	$\sqrt[12]{2}$	$\sqrt[6]{2}$	$\sqrt[3]{2}$	$\sqrt{2}$	$(\sqrt[1.5]{2})$	$(\sqrt[1.2]{2})$	$\sqrt[1]{2}$
$\sqrt[E]{10}$	$\sqrt[40]{10}$	$\sqrt[20]{10}$	$\sqrt[10]{10}$	$(\sqrt[20/3]{10})$	$\sqrt[5]{10}$	$\sqrt[4]{10}$	$(\sqrt[20/6]{10})$
$A=\frac{\varphi-1}{\varphi}100\%$	~ 5	10	20	30	40	45	50

Series of geometrical structure, and the same values of the constant ratios are used for the numbers of full strokes per minute (back and forth) of machine tools with reciprocating primary cutting motions and for feed series. It becomes necessary to resort to other than geometrical series when their use is prevented by certain features of the mechanism employed to change the number of strokes or rates of feed (ratchet mechanism in the feed drive). This may also be necessary in complying with certain requirements such as the provision of a feed series for cutting threads in a pitch range.

2-4. Choosing the Ratios for the Series of Spindle Speeds, Numbers of Full Strokes and Feeds

After determining the maximum and minimum spindle speeds (or numbers of full strokes per min) n_1 and n_z, and consequently, the range ratio $R_n = \frac{n_z}{n_1}$, for the machine tool being designed, it is necessary to establish z, the number of speed steps (which is the same as choosing the ratio φ). The number of speed steps is related to the range ratio R_n and ratio φ by equation (34) from which it is evident that at a given R_n value, the number of steps increases rapidly with a reduction in φ (Fig. 5). Thus, in selecting φ and z it is necessary to find an economically expedient compromise between the effort to reduce losses in cutting speed by making more steps and thereby complicating the construction, and the effort to reduce the cost of the machine tool by keeping its construction as simple as possible. A final decision should take into consideration the following:

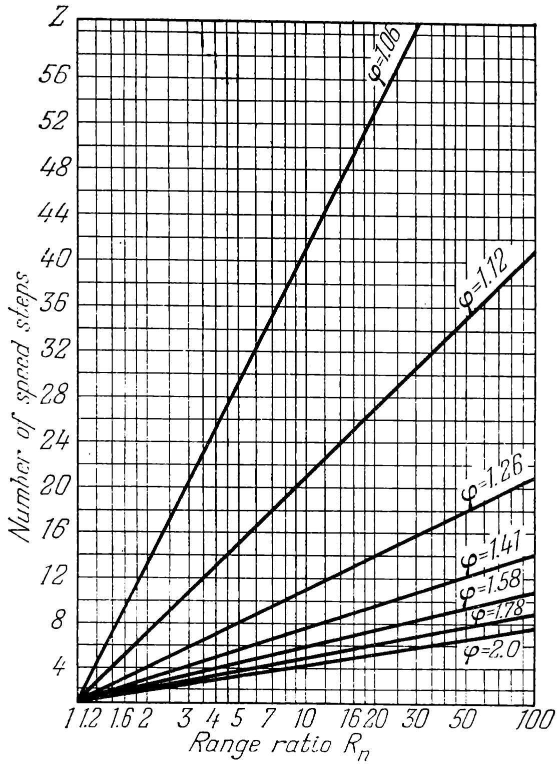


Fig. 5. Relationship between the speed range ratio, number of speed steps and the progression ratio of the speed series

1. In the great majority of general-purpose machine tools with stepped speed variation, the use of a ratio $\varphi = 1.26$ or 1.41 leads to quite satisfactory operation.

2. If speed changes are to be made in the drive gear train by means of change gears, then the value $\varphi = 1.12$ or 1.26 proves satisfactory for machine tools intended for lot or mass production (automatic and semiautomatic machine tools).

3. A geometrical series has an excessively large number of steps in the high speed range (used mainly in machining small diameters). Indeed, the maximum diameter interval x_j, accommodated at a constant cutting speed by two adjacent members (steps) of the speed series (Fig. 6), is

$$x_j = d_{j-1} - d_j = \frac{v}{\pi n_{j-1}} - \frac{v}{\pi n_j} = \frac{v}{\pi n_{j-1}}\left(1 - \frac{n_{j-1}}{n_j}\right) = d_{j-1}\left(1 - \frac{1}{\varphi}\right) = d_{j-1}A \qquad (38)$$

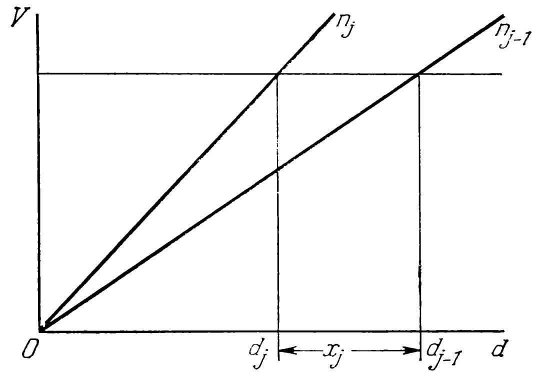


Fig. 6. Diagram referring to equation (38)

If, to avoid a large number of steps in the series, the diameter interval x_j is taken so that it is not less than the interval Δd of standard bar stock diameters or of tool sizes (for instance, drills), then

$$x_j = d_{j-1}A \geqslant \Delta d$$

from which

$$d_{j-1} \geqslant \frac{\Delta d}{A} = \frac{\varphi}{\varphi - 1} \Delta d \tag{39}$$

This last equation enables us to find the minimum diameters of bar stock to be machined and of tools to be used in machine tools with various φ values and, conversely, the minimum values of φ that can be assigned to machine tools with various minimum diameters of work to be machined. Therefore, large ratios ($\varphi = 1.58$, and sometimes $\varphi = 1.78$) are used in small machine tools which accommodate small work diameters, while smaller values ($\varphi = 1.26$, $\varphi = 1.12$, and sometimes $\varphi = 1.06$) are used in heavy machine tools.

4. It is good practice to select a number of speed steps z having the factors 2 and 3, so that

$$z = 2^{E_1} 3^{E_2} \tag{40}$$

where E_1 and E_2 are whole numbers.

This requirement is met by the values: $z = 2, 3, 4, 6, 8, 9, 12, 16, 18, 24, 27, 32$ and 36. The most frequently used values are

$$z = 3, 4, 6, 8, 12, 18 \text{ and } 24 \tag{41}$$

The reasons for making this requirement are indicated in Chap. 3; they are not valid when speeds are changed by means of change gears.

Machine tool industry standard H11-1 recommends the ratios $\varphi = 1.26$, 1.41 and 1.58 for speed and feed series in designing machine tools.

The range ratios of spindle speed variation R_n and feed variation R_s, and the number of steps of main drive speeds z and of feeds z_s may vary within quite large limits. For each type and size of machine tool the values of these specifications depend upon the purpose of the machine tool, the nature of the manufacturing process, the type of cutting tools to be used, and especially, upon the required degree of versatility. The more versatile the new machine tool is to be, the more different types of tools that are to be used (carbide-tipped, high-speed steel, and ceramic tools), the larger the range ratios R_n and R_s must be for efficient operation. The influence of these factors can be demonstrated by the fact that in cylindrical grinding machines, for example, in which $v \cong$ const and the wheel diameter varies within the limits $R_d \leqslant 2$, the wheel spindle speed range ratio is $R_n \leqslant 2$, while in horizontal boring machines the range ratio of feed variation is, on the contrary, very large, reaching values of $R_s \cong 1000$ in certain models, and sometimes more.

In most cases, $z \leqslant 36$ in machine tools with a rotary primary cutting motion if a variable-speed drive (with infinitely variable speed variation) has not been used. The same is true for the number of feed steps ($z_s \leqslant 36$); engine lathes designed for cutting threads of various pitches have feed mechanisms with $z_s = 48$ to 60 and more.

The range ratios of the number of strokes per minute R_n or of the working stroke speeds R_v of machine tools with a reciprocating primary cutting motion are narrower than the range ratios R_n in machine tools with a rotary primary motion.

The values of R_n, R_s, z and z_s should be much smaller in specialized and, in particular, special machine tools, than in general-purpose models.

2-5. Determining the Power Rating of the Electric Motor

It is often very difficult to determine the power ratings of the electric motors of a new machine tool. This is due chiefly to the lack of sufficient data as yet on such factors as: (1) the laws governing the cutting and feed forces in various chip-removal processes, especially during transient operation (starting and reversing); (2) conditions of machine tool operation, especially general-purpose models; and (3) friction losses in the drive, especially at high rotational speeds. Hence, the useful power of the drive and the required power rating of the electric motor cannot always be established with sufficient accuracy by calculations. In some cases, the rating must be determined experimentally or by analogy with the power rating of existing machine tools.

The *required power of the main drive* is determined on the basis of the useful power, calculated for the most effective cutting conditions. Useful power is calculated for the given operations in designing special machine tools, and for several workpieces, typical of those that are to be machined, in designing single-purpose models. In designing general-purpose machine tools, the useful power is calculated for the maximum cutting speeds and feeds. Since the operating conditions of general-purpose machine tools of the same model may differ greatly in different manufacturing plants, the required power is also determined by correlation with the power ratings of several machine tools of up-to-date construction and near to the machine tool being designed in type and size features.

The required power rating N_{em} of the electric motor is determined from the established useful power N by the relationship

$$N_{em} = N + N_f = N + N_{nl} + N_a \tag{42}$$

where N_f is the power lost in overcoming friction. N_f is the sum of the constant no-load power N_{nl} which under load is equal to the constant part of the whole losses, not depending upon the load, and the power N_a which represents the additional losses that depend upon the load (loading losses).

The total efficiency of the main drive is

$$\eta = \frac{N}{N_{em}} \tag{43}$$

and if η is known, the power of the electric motor can be determined from the equation

$$N_{em} = \frac{N}{\eta} \tag{44}$$

The efficiency η varies with the useful load, speed, kinematic arrangement of the drive, construction of its elements and the quality of their manufacture. The influence of some of these factors can be seen in Fig. 7 which shows experimental values of η obtained by G. Levit for the drive of turret lathe, model 1M36, as a function of load, spindle speed and the speed of the input shaft in the speed gearbox.

To determine the required power rating of the drive motor, it is sufficient to know the efficiency η corresponding to full effective load on the kinematic chain. In the absence of experimental data, a tentative estimate of η may lead to errors which are especially large in the case of high-speed machine tools ($n_{spind} \geqslant 1000$ rpm as the design speed). The total efficiency η may range from 0.7 to 0.85 for machine tools with a rotary primary motion having a single-motor drive.

A rough estimation of the efficiency of a machine tool drive can be avoided if the efficiency is arbitrarily defined as the product

$$\eta = \eta_1 \eta_2 \eta_3 \ldots = \prod_j \eta_j \tag{45}$$

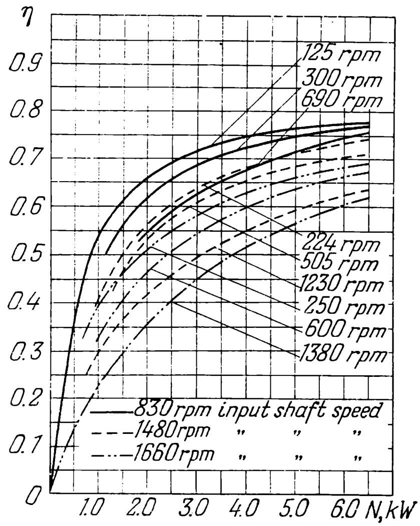


Fig. 7. Experimental values of the efficiency of the drive in a turret lathe vs the load for various speeds of the spindle and input shaft of the speed gearbox (after G. Levit)

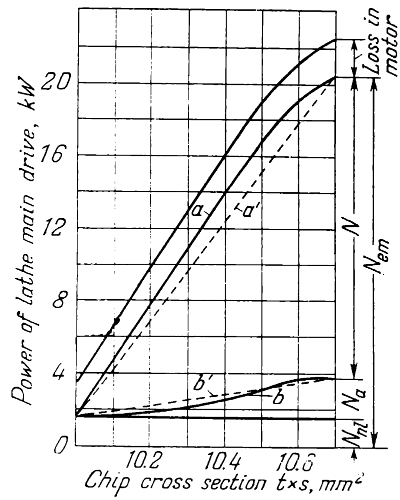


Fig. 8. Power balance diagram of a lathe main drive (power developed by the drive vs chip cross section)

where η_j are particular efficiency values of the separate elements and transmissions making up the kinematic chain of the drive. The values η_j can be taken from the data in the study-course *Machine Design* where they are given for the full design load of the transmissions.

In operation at the same power but at higher speeds of the spindle and intermediate shafts than for the design train of transmissions, the elements of the drive will operate at underload conditions in comparison to the rated load permitted for increased rotational speeds. In this case, equation $\eta' = \prod_j \eta_j$ will give excessively large values of the efficiency.

The curves in Fig. 8 were plotted from experimental data and represent the power balance diagram of a lathe main drive for various chip cross sections. If all the curves in the diagram were straight lines (straight dash lines a' and b'), we could write

$$\frac{N}{N_{em}-N_{nl}}=\frac{N}{N+N_a}=\eta''=\text{const} \tag{46}$$

if by efficiency η'' we take into account only loading losses, but ignore the no-load losses.

Hence

$$N_{em} = \frac{N}{\eta''} + N_{nl} \tag{47}$$

According to experimental data $\eta'' = 0.88$ to 0.90. The no-load power N_{nl} of gear drives, not dependent upon the load, can be determined by means of an empirical formula proposed by G. Levit (see p. 75).

Formula (47) enables the power rating N_{em} to be determined, not only for various spindle speeds, but also at various values of the useful power N, i.e., at conditions when the spindle drive is underloaded.

If the motor is to power several kinematic chains of the machine tool, then its rating is

$$N_{em} = \sum_{i} \frac{N_i}{\eta_i''} + N_{nl} \tag{48}$$

where N_i = effective power required by the final member of any one of the kinematic chains

η_i'' = efficiency of this chain, taken equal to 0.88-0.90 or determined by experiments.

If the right-hand side of the last equation refers to the power requirement of the feed drive in lathes, drilling machines or grinders, it is so small that it can be neglected.

The power requirement of a feed drive can be calculated from the feed force Q and rate of feed v_s, taking into account the efficiency, which is seldom very large for feed drives (of the order of 0.15 to 0.20, and sometimes even less). Feed forces can be calculated by the following practical formulas, recommended by machine tool industry standard H48-61 and derived by D. Reshetov and G. Levit:

for the saddles of lathes with vee or combined ways

$$Q = kP_x + f'(P_z + G) \tag{49}$$

for the saddles of engine and turret lathes, and the tables of milling machines with flat ways

$$Q = kP_x + f'(P_z + P_y + G) \tag{50}$$

for the tables of milling machines with dovetail ways

$$Q = kP_x + f'(P_z + 2P_y + G) \tag{51}$$

and for the spindles of drilling machines

$$Q = (1 + 0.5f)\,P_x + f\frac{2M_t}{d} \cong P_x + f\frac{2M_t}{d} \tag{52}$$

where P_x, P_y and P_z = components of the cutting force
G = weight of the parts being traversed
M_t = torque on the spindle
d = diameter of the spindle
f' = coefficient of friction in the ways
f = coefficient of friction between the spindle quill and its seat in the spindle head, and in the spline fittings or keys and keyways of the spindle
k = factor taking into account the influence of the overturning moment.

The following values can be taken, assuming normal lubrication: $k = 1.15$ and $f' = 0.15$ to 0.18 for lathes with vee or combined (vee and flat) ways; $k = 1.1$ and $f' = 0.15$ for engine and turret lathes with flat ways; $k = 1.4$ and $f' = 0.2$ for the tables of milling machines; and $f = 0.15$ for the quills of drilling machines.

CHAPTER 3

DEVELOPING THE KINEMATIC SCHEME OF A MACHINE TOOL

3-1. Determining the Transmission Ratios of the Mechanisms in the Kinematic Chain

Principal Kinematic Relationships in the Spindle Drive

The kinematic chain of transmission in the spindle drive should provide a gradation of spindle speeds n in a geometrical series (progression) with the selected progression ratio φ and the given maximum and minimum speeds $n_{min} = n_1$ and $n_{max} = n_z$. Methods for solving these problems are based on kinematic calculations.

Any regularity in the series of speeds n is the result of a similar regularity in the series of transmission ratios i in the drive.

If spindle speeds are obtained by means of only a single transmission group, i.e., by making engagements between sets of simple trains arranged on two shafts, any series of spindle speeds can be produced by selecting the corresponding series of transmission ratios for the trains of the group.

However, if the different spindle speeds are obtained by consecutive engagement of transmission groups, the drive being through a compound train, only a geometrical series of speeds can be set up (Fig. 9).

This method of speed changing enables: (1) the number of spindle speed steps to be increased, (2) the range of drive variation to be extended and (3) the number of simple trains, required to obtain the speeds, to be reduced.

These advantages, inherent in the design of a geometrical series in addition

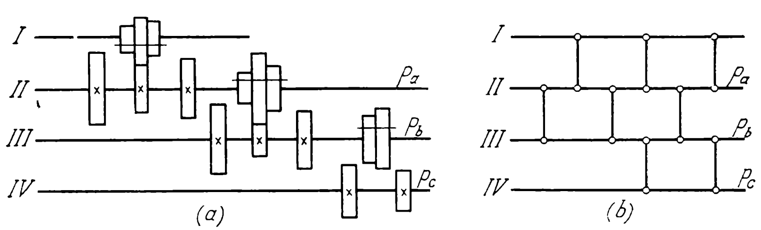


Fig. 9. A drive with consecutive engagement of transmission groups: (a) kinematic diagram; (b) structural diagram

to its economical advantages, have made this series of spindle speeds the principal one used in machine tool engineering.

1. *Number of speed steps.* In obtaining different speeds by consecutive engagements of transmission groups (Fig. 9), the number of simple trains in each consecutive group being denoted by p_a, p_b, p_c . . ., the number of spindle speeds z is equal to

$$z = p_a p_b p_c \dots p_r \tag{53}$$

For example, for the drive arrangement shown in Fig. 9

$$z = p_a p_b p_c = 3 \times 3 \times 2 = 18$$

2. *Range ratio of speed variation in a drive.* When consecutive chains of transmission are engaged, the total transmission ratio of the drive is equal to the product of the transmission ratios of the simple trains that make up the drive. Thus, applying this principle to calculate the maximum and minimum transmission ratios, i_{max} and i_{min}, of the drive, we obtain

$$i_{max} = i_{a\,max} i_{b\,max} i_{c\,max} \dots i_{r\,max} \tag{54}$$

and

$$i_{min} = i_{a\,min} i_{b\,min} i_{c\,min} \dots i_{r\,min} \tag{55}$$

where the subindex a refers to p_a, subindex b to p_b, etc.

From this it follows that the range ratio of the drive is

$$R_n = \frac{n_{max}}{n_{min}} = \frac{i_{max}}{i_{min}} = R_a R_b R_c \dots R_r \tag{56}$$

where $R_a = \frac{i_{a\,max}}{i_{a\,min}}$, and similarly, R_b and R_c are the range ratios of the transmission groups.

3. *Setup equation of the drive.* The possibility of using consecutively engaged multiplier transmission groups for changing speeds is a most important property possessed only by geometrical series of speeds. Therefore, the kinematic conditions for changing the speeds of such drives are governed by the kinematic properties of the multiplier transmission groups.

To reveal these general properties of such groups, let us assume that a gearbox with a range ratio of speed variation R_{gb} is linked into a train of simple constant transmissions (Fig. 10) so that the spindle speeds can be changed in a geometrical series within this range from n_1 to n_k.

Next, we add a series of transmissions (*2*, *3*, . . ., p) to one of the simple transmissions (transmission *1* in Fig. 11) to extend the series of spindle speeds, thereby forming a multiplier transmission group with the ratios i_1, i_2, i_3, . . ., i_p. When the transmission with the ratio i_1 is engaged, the

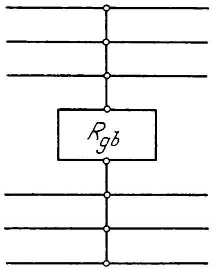


Fig. 10. Structural diagram of a drive with the range ratio R_{gb}

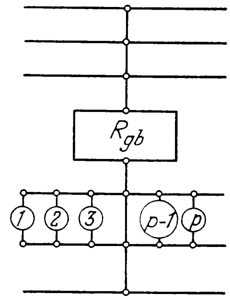


Fig. 11. Introduction of a multiplier group of p transmissions for extending the range ratio of the drive beyond R_{gb}

speed gearbox can change the spindle speeds n_j in the geometrical series

$$n_1, \; n_2, \; \ldots, \; n_{k-1}, \; n_k \tag{57}$$

To extend this series of speeds, we change over the multiplier group from transmission *1* to transmission *2*, after which we can obtain the following members of the geometrical speed series:

$$n_{k+1}, \; n_{k+2}, \; \ldots, \; n_{2k} \tag{58}$$

Hence

$$\frac{i_2}{i_1} = \frac{n_{2k}}{n_k} = \frac{n_{2k-1}}{n_{k-1}} = \ldots = \frac{n_{k+1}}{n_1} = \frac{n_k \varphi}{n_1} = R_{gb}\varphi \tag{59}$$

Thus, to further increase the spindle speed in a geometrical series, a new engagement is made in the multiplier group only after utilizing all the speed changes available in the gearbox with the range ratio R_{gb}. This requires the following relationship between the ratios of the transmissions in the multiplier group:

$$\begin{aligned} i_1 : i_2 : i_3 : \ldots : i_p &= n_1 : n_{k+1} : n_{2k+1} : n_{3k+1} : \ldots : n_{(p-1)k+1} = \\ &= n_1 : n_1 R_{gb}\varphi : n_1 (R_{gb}\varphi)^2 : n_1 (R_{gb}\varphi)^3 : \ldots : n_1 (R_{gb}\varphi)^{p-1} \end{aligned} \tag{60}$$

These relationships are shown in Fig. 12 where the speeds are plotted on a logarithmic scale. The intervals between the lines indicating adjacent speeds are equal to $\log \varphi$, while the interval between the lines n_k and n_1, equal to $\log n_k - \log n_1 = \log \frac{n_k}{n_1} = \log R_{gb}$, is the range ratio of the gearbox. Hence, the ratios of the transmissions in the multiplier group make up a geometrical series of transmissions with the progression ratio $\varphi_p = \varphi R_{gb}$. Thus

$$i_1 : i_2 : \ldots : i_p = 1 : \varphi R_{gb} : (\varphi R_{gb})^2 : \ldots : (\varphi R_{gb})^{p-1} \tag{61}$$

where R_{gb} is the range ratio of the whole complex of transmissions, in reference to which the given group is a multiplier one and therefore the consecutive one in the kinematic order in which the groups are arranged. Since, each group of transmissions is a multiplier one in reference to the whole complex of transmission groups that precede it kinematically, equation (61) expresses the main general law for setting up all group transmissions in the spindle drive.

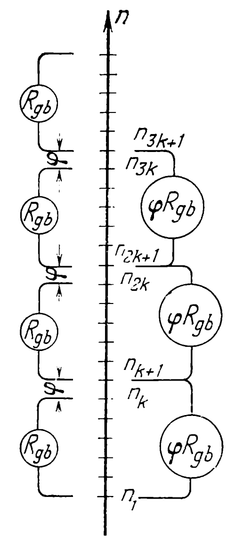


Fig. 12. Diagram referring to equation (60)

4. *Characteristic of a transmission group.* The progression ratio of the series of transmission ratios in a transmission group can be expressed as

$$\varphi_p = R_{gb}\varphi = \varphi^{z_k-1}\varphi = \varphi^{z_k} = \varphi^x \tag{62}$$

where z_k is the number of speed steps in the whole complex of transmissions with a range ratio R_{gb} kinematically preceding the given group. The exponent x is called the *characteristic of the group.*

In this manner, the characteristic of a group is equal to the number of speed steps of the whole complex of transmission groups kinematically preceding the given group.

The general setup equation for group transmissions (61) can be written as

$$i_1 : i_2 : i_3 : \ldots : i_p = 1 : \varphi^x : \varphi^{2x} : \ldots : \varphi^{(p-1)x} \tag{63}$$

The first in the kinematic order of group arrangement—the so-called *main group*—is a multiplier group in relation to the whole complex of simple transmissions, giving $z_k = 1$, and consequently $x_1 = z_k = 1$.

In the second group of transmissions (in the same sense)—the so-called *first extension group*—$z_k = p_1$ and $x_2 = p_1$, where p_1 is the number of transmissions in the main group.

In the third group of transmissions—the *second extension group*—$z_k = p_2$ and $x_3 = p_1p_2$, where p_2 is the number of transmissions in the first extension group, etc.

Equations (61) and (63) can be used to find the ratios of all the transmissions in a group if the ratio i of one transmission is known.

5. *Formula for the structure of the drive.* To apply the setup equation (63), it is necessary first to determine the characteristics of all the groups and, therefore, the place of each group in the kinematic order of arrangement and the number of transmissions in the groups which precede it kinematically. Therefore, in formula (53) for determining the number z of speed steps, it proves convenient to denote the subindex of p by the number of the group in the kinematic order of arrangement and to arrange the groups in this for-

mula in the same order as they are actually arranged along the transmission train from the motor to the spindle. With this notation, formula (53) is converted into the formula for the structure of the drive.

Analytical Method of Determining Transmission Ratios

The initial data in kinematic calculations of a spindle drive are: series of spindle speeds n with a definite progression ratio φ of the series of transmission ratios and a definite number z of speed steps from $n_{min} = n_1$ to $n_{max} = n_z$, and the speed n_{em} of the electric motor.

In accordance with these initial data, the following are set down tentatively: formula (53) for the structure of the drive; number of simple transmissions required for the construction of the drive; and their arrangement among the group transmissions. Then a kinematic diagram of the drive is drawn and used as the basis for calculations during which it may be necessary to make corrections in the diagram.

Standard transmission ratio. Standard speeds (according to machine tool industry standard H11-1) should be assigned, wherever possible, to all shafts of the drive. Since all standard speed series are contained in the finest series (with $\varphi = 1.06$), then, in the general form, the standard transmission ratio of any transmission in the drive can be expressed as

$$i_{st} = 1.06^{\pm E} \tag{64}$$

where E is a whole number.

Calculations are simplified if all transmission ratios are expressed in terms of the progression ratio φ of the series of spindle speeds in the drive being designed.

Limiting transmission ratios. To avoid excessively large diameters of driven gears and a consequent increase in the radial overall dimensions of the drive, it is general practice to limit the transmission ratio of gears in a gearbox by the value $i_{min\ lim} = \frac{1}{4}$. The maximum transmission ratio assigned is $i_{max\ lim} = \frac{2}{1}$ for spur gearing and $i_{max\ lim} = \frac{2.5}{1}$ for helical gearing. A value $i_{max\ lim} = \frac{4}{1}$ may be permissible in small machine tools in case of smooth rotation of the driving shaft (electric motor shaft or a shaft driven through a flexible coupling from the electric motor).

The accepted range for feed gearboxes (with slow gearing and small-diameter gears) is $\frac{1}{5} \leqslant i \leqslant \frac{2.8}{1}$.

Thus, the limiting maximum range ratio in a two-shaft group transmission is

$$R_{lim} = \frac{i_{max\ lim}}{i_{min\ lim}} = 8 \tag{65}$$

If it is necessary to obtain a larger range ratio in a group transmission (10 or 12 as an extreme case), extension devices with consecutive engagement of reduction transmissions are used.

The *ratios of the transmissions in the drive* are determined as follows:

1. The characteristics of the group transmissions are calculated by formula (53) for the structure of group transmissions.

2. Using setup equation (63), the relationship is determined for each group of transmissions between the transmission ratios of the group transmissions so as to obtain a gradation of spindle speeds in accordance with the given geometrical series.

3. The minimum transmission ratio i_{min} for the whole drive is determined. It is expressed in the form of the exponent of the progression ratio φ of the series of spindle speeds. Thus

$$i_{min} = \frac{n_1}{n_{em}} = \frac{1}{\varphi^q} \tag{66}$$

Exponent q is taken from the table H11-1 which lists standard series of numbers to be used in machine tool engineering.

4. Taking into consideration the values $i_{min\ lim}$ and $i_{max\ lim}$, as well as the features of the various simple and group transmissions, the transmission ratios of the simple transmissions and the minimum transmission ratios of the group transmissions are assigned, in accordance with equation (55), in such a manner that their product is equal to i_{min} for the whole drive. To this end, all the transmission ratios are expressed in the form $i = \varphi^{\pm U}$ so that in equation (55) the algebraic sum of the exponents U is equal to q.

5. Having thus obtained the values $i_1 = i_{min}$ for all group transmissions, the values i for the other transmissions of each group are found by the use of equation (63).

Semigraphical Method of Determining Transmission Ratios

It is evident from the foregoing that the transmission ratios, their gradation and the speeds of all the shafts in the drive can be expressed in the form of powers of the progression ratio φ for the series of spindle speeds. Therefore the kinematic linkages of the drive can be conveniently depicted graphically on logarithmic scales with a constant interval between adjacent points of the scale equal to log φ (scale division value).

Structural diagram. Formula (53) for the structure of group transmissions and setup formula (63) are represented in a special graph called the structural diagram.

To construct a structural diagram, a series of parallel straight lines are drawn, horizontally for example, at intervals equal to log φ and of a number equal to the number of spindle speed steps z. Then, a series of vertical lines are drawn at random (approximately equal) distances from each other (Fig. 13). Each zone between two adjacent vertical lines is allotted to a transmission group in the order of the arrangement of the groups along the train of transmission. This has been done in Fig. 13 for a drive whose design is shown schematically in Fig. 14 and whose structural formula is

$$z = p_a p_b p_c = p_1 p_2 p_3 = 3_1 \times 3_2 \times 2_3$$

Using the structural formula, we calculate the characteristics x_j of all the groups and write them above the zones of the corresponding groups. On the left vertical line of the first zone (in the order of the arrangement in the drive) we plot a point located symmetrically in respect to the horizontal lines (point O in Fig. 13). Opposite this point, on the right vertical line of this first zone, we plot symmetrically as many points as there are transmissions in this group. The distance between these points is equal to the characteristic of the group expressed as a multiple of log φ. These points are then connected by straight rays with point O on the left vertical line of this group zone.

In the zone of the second group, as many rays are drawn symmetrically from each point on the left vertical line of this zone as there are transmissions in this group, the distance between the ends of the rays being equal to x_2 log φ intervals. In the given example $x_2 = 3$. Likewise, two rays ($p_3 = 2$) are drawn symmetrically from each point on the left vertical line of the zone for the last group, the distance between the rays being equal to 9 log φ intervals ($x_3 = p_1 p_2 = 3 \times 3$).

As is evident from this construction, the structural diagram contains the following data concerning the drive: number of transmission groups; number of transmissions in each group; the relative order of the groups in the train of transmissions; the order of kinematic arrangement of the groups, i.e., their characteristics and the relation between the transmission ratios; the range ratio of each transmission group and the drive as a whole; and the number of speed steps of each shaft in the drive.

All kinematic relationships in the structural diagram are expressed by the powers of the progression ratio φ.

The construction of the structural diagram is equivalent to the application of equations (53), (56) and (63).

The structural diagram indicates the relationships between the transmission ratios of the group transmissions but does not give the actual values.

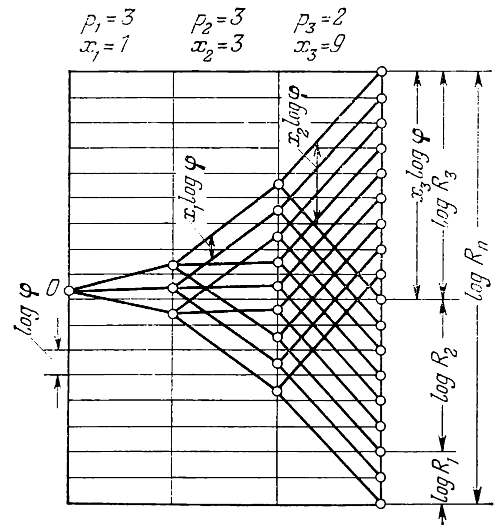


Fig. 13. Structural diagram of the drive shown in Fig. 14

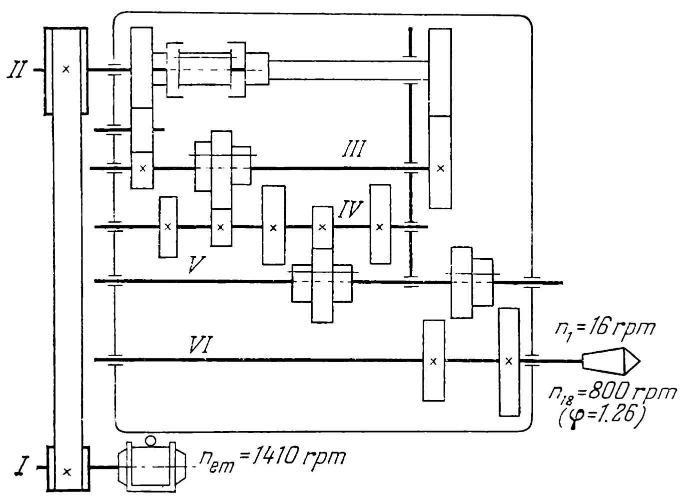


Fig. 14. Kinematic diagram of a drive

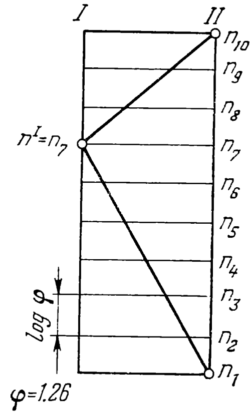


Fig. 15. Constructing the speed chart for a transmission group

Hence, each structural diagram represents a whole series of actual drives shown in their general form.

Speed chart. Concrete values of the transmission ratios for all the transmissions in the drive and speeds of all the shafts are determined by constructing the speed chart. This is done on the basis of the kinematic diagram of the drive. Each shaft is represented by a vertical straight line in the chart. Horizontal straight lines spaced at equal intervals (equal to log φ) are marked with all the speeds of the corresponding shaft in the limits from the minimum to the maximum speed.

Transmissions, engaged at definite speeds of the driving *I* and driven *II* shafts (Fig. 15), are shown on the chart by rays connecting the points on the shaft lines representing these speeds.

The transmission ratio is expressed in the form φ^m, where m is the number of intervals between the horizontal lines spanned by the corresponding ray. If the speeds are written from the bottom to the top in the increasing order of magnitude, then for a speed (increase) transmission, i.e., $i > 1$ and $m > 0$, the ray is inclined upward (in the direction from the driving to the driven shaft). In the case of a reduction transmission, i.e., $i < 1$ and $m < 0$, the ray is inclined downward. For a transmission where $i = 1$, the exponent $m = 0$ and the ray is horizontal. Thus, for the transmission engaged at $n^{\mathrm{I}} = n_7$ and $n^{\mathrm{II}} = n_{10}$ (see Fig. 15), the ray is inclined upward and spans three intervals so that the transmission ratio is $i^{\mathrm{I/II}} = \frac{n_{10}}{n_7} = \varphi^3$. In the transmission engaged at $n^{\mathrm{I}} = n_7$ and $n^{\mathrm{II}} = n_1$, the ray spans six intervals and is inclined downward. Thus $i^{\mathrm{I/II}} = \frac{n_1}{n_7} = \frac{1}{\varphi^6}$.

The speed chart whose construction is shown in Figs. 16 and 17 refers to the lathe spindle drive with $\varphi = 1.26$ $(=\sqrt[3]{2})$ whose kinematic diagram is shown in Fig. 14.

In accordance with the kinematic diagram of this drive, we draw six vertical lines, denoted by the Roman figures *I* through *VI*, in the same manner as the shafts in Fig. 14. Taking into consideration the specific features of various transmissions and the limiting values of the transmission ratios, $i_{min\ lim} = \frac{1}{4} = \frac{1}{\varphi^6}$ and $i_{max\ lim} = \frac{2}{1} = \varphi^3$, we draw the train of transmissions for reducing the speed from $n^{\mathrm{I}} = 1410$ rpm to $n_1^{\mathrm{VI}} = 16$ rpm (Fig. 16).

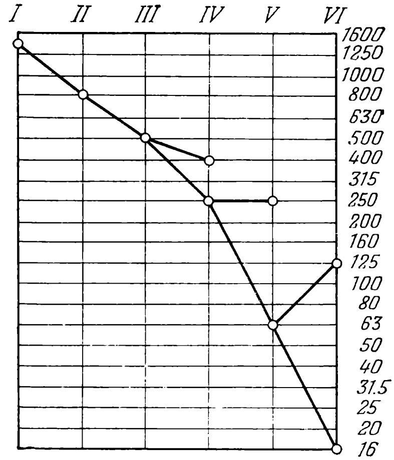


Fig. 16. Constructing the speed chart for the drive shown in Fig. 14

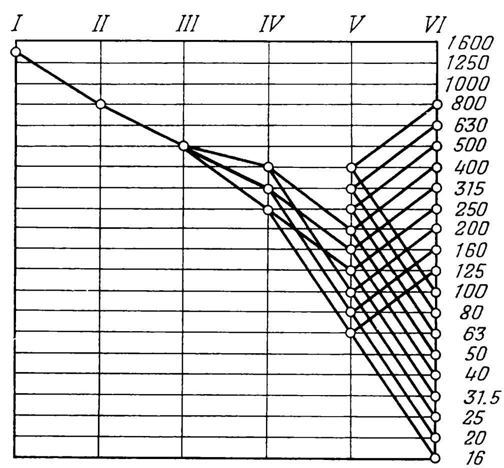


Fig. 17. Finished speed chart for the drive shown in Fig. 14

Next, we draw the rays for the transmissions i_{max} of group transmissions, using the range ratios represented in the structural diagram (Fig. 13) by the numbers of log φ intervals. All of these rays are located within the limits $i_{max\ lim} = 2 = \varphi^3$. In selecting the value $i_{min}^{V/VI} > \frac{1}{\varphi^6}$ (i.e., if the ray of this transmission spanned less than six intervals) the ray of transmission $i_{max}^{V/VI}$ would then span more than three intervals, so that $i_{max}^{V/VI} > i_{max\ lim} = 2$, which would be undesirable (see p. 44).

The structural diagram is employed to complete the speed chart. This is done by superimposing the zone of each group with the corresponding zone of the chart so that the lower ray of the network coincides with the ray i_{min} of the transmission train, but with the distances between the ends of the rays remaining the same as in the structural diagram (compare Figs. 17 and 13).

Thus constructed, the chart represents the engagements of all the group transmissions for all speeds of the spindle (and of the shafts in the drive) with constant relationships between the transmission ratios required to obtain a geometrical series of spindle speeds.

A speed chart contains all the data of a structural diagram and, in addition, it reveals the number of simple transmissions required for the design layout of the drive and for reducing the speed from that of the motor to that of the spindle. It also shows the relative position of the simple transmissions in reference to the group transmissions, the ratios of all the transmissions and of the whole drive at all spindle speeds, and the speeds of all shafts of the mechanism for all the possible engagements of the group transmissions.

Thus, the speed chart contains the structure of the drive, all kinematic data expressed in terms of the progression ratio φ of the series in a comprehensive and convenient form. These constitute the great advantages of the semigraphical method of calculation in machine tool design.

The analytical method of kinematic calculations is employed in research and for tentative calculations in studying the different possible versions of the drive.

The time required by calculations using this method can be substantially reduced by efficient application of standard H11-1 to express the range ratios and transmission ratios in terms of powers of progression ratio φ.

3-2. Selecting the Transmission Ratios for Drives Powered by a Multiple-Speed Electric Motor

Multiple-speed squirrel-cage induction motors and shunt-wound d-c motors find application in drives with stepped spindle speed variation. The latter are frequently used with a generator-motor (adjustable-potential or Ward-Leonard) system mainly in heavy machine tools. The use of a multiple-speed

electric motor enables the mechanical part of the drive to be simplified. This simplification, however, does not always economically justify the use of more expensive multiple-speed motors.

The possibility of changing speeds while the machine tool is running is a great advantage of multiple-speed motors. Consequently, they are often used in the drives of small machine tools in conjunction with transmissions that can also be rapidly changed over without stopping the machine tool (gearing in which speeds are changed by the engagements of mechanical, electromagnetic or hydraulic friction clutches, variable-speed transmissions, etc.). The aim of applying such arrangements is to reduce the handling time when the machining time is very short, and also to automatically change spindle speeds and rates of feed during the working cycle in automatic machine tools of various sizes.

Insofar as setting up a drive to a geometrical series of speeds is concerned, a multiple-speed electric motor with stepped speed variation can be regarded as a group (electrical group) of p_{em} transmissions with a progression ratio of the series of transmission ratios $\varphi_p = \varphi_{em}$.

Induction motors, for whose synchronous speeds $\varphi_{em} \neq$ const, do not permit a general solution when combined with a speed gearbox and are not to be considered here. The following equation must hold true [see equation (62)] in order to obtain a geometrical series of speeds:

$$\varphi_p = \varphi_{em} = \varphi^{z_k} = \varphi^{x_{em}}$$

from which

$$z_k = \frac{\log \varphi_{em}}{\log \varphi} \tag{67}$$

where z_k is the number of speed steps in the whole complex of transmissions in the groups preceding the electrical group in the kinematic order of group arrangement.

If $z_k = 1$ and $\varphi_{em} = \varphi$, the electrical group is the main one. This is the case for shunt-wound d-c motors when p_{em} positions of the adjusting device (p_{em} contacts of the speed-adjusting rheostat) provide a gradation of motor speeds with $\varphi_p = 1.12 = \varphi$. In this case, $p_{em} = p_1 = 11$ and $R_{em} = 1.12^{10} = 3.15$, and the motor is combined with the mechanical part of the drive as usual on the basis of equations (53), (63) and (67).

For an induction motor with $\varphi_p = \varphi_{em} = 2$,

$$z_k = x_{em} = \frac{\log 2}{\log \varphi} \tag{68}$$

In order that $z_k = 1$ and the electrical group be the main group, it is necessary that $\varphi = 2$. This is seldom the case in actual practice.

At the values $\varphi = 1.26$ and $\varphi = 1.41$, the last equation gives $z_k = x_{em} = 3$ and 2, respectively. In these cases, the electrical group serves as the

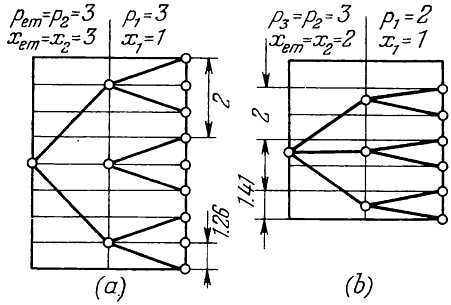


Fig. 18. Structural diagram of a drive with a three-speed electric motor serving as the first extension group:
(a) at $\varphi = 1.26$; (b) at $\varphi = 1.41$

first extension group (as is most frequently the case) and the main group should have $p_1 = 3$ for $\varphi = 1.26$ or $p_1 = 2$ for $\varphi = 1.41$ (Fig. 18*a* and *b*).

If z_k obtained from equation (68) is a value that can be expanded into two factors (for example, $z_k = x_{em} = 4$ or 6 for $\varphi = \sqrt[4]{2} = 1.19$ or $\varphi = \sqrt[6]{2} = 1.12$) two solutions are possible, namely $z_k = x_{em} = p_1$ for the preceding case, and $z_k = x_{em} = p_1 p_2$ in which the electric motor is the second extension group.

The kinematic possibilities offered by the application of multiple-speed induction motors can be substantially extended by using nonuniform groups. Such groups are formed by combining two groups that are not adjacent in the kinematic order of arrangement—the second extension and the main groups.

The combined group (see p. 57) is shown by dashed rays in Fig. 19*a* and *b*.

In the initial uniform structure, the electrical group may be the first extension group with $p_{em} = p_2 = 2_2$ if it is arranged as the first group in the design. In this case, definite distances between the ends of the rays can be established for the nonuniform group (Fig. 20*a* and *b*).

If a nonuniform group is combined with a multiple-speed electric motor (Fig. 21*a* and *b*), the structure of the drive is considerably simplified (especially in small high-speed machine tools requiring a small degree of speed reduction). Thus, in place of the three groups in the structure shown in Fig. 19*a* and *b*, only one group of mechanical transmissions is required.

Structures Deviating from Normal Uniform Structure

It follows from equation (62) that for the last extension group, consisting of p_m transmissions, the characteristic is $x_m = z_k = p_1 p_2 p_3 \ldots p_{m-1}$.

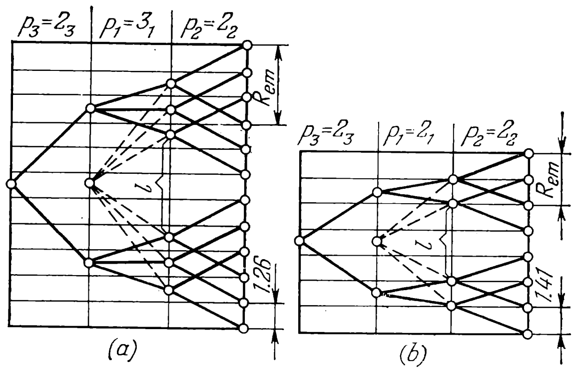


Fig. 19. Constructing the zone of a nonuniform group:
(a) for a group of six transmissions; (b) for a group of four transmissions

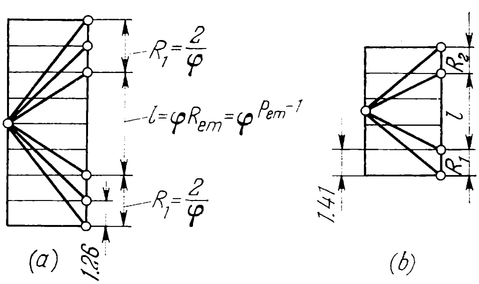


Fig. 20. Zone of a nonuniform group:
(a) of six transmissions; (b) of four transmissions and with a two-speed electric motor as the first extension group

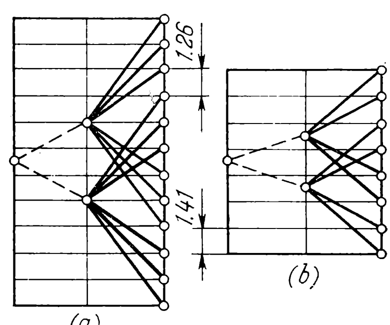


Fig. 21. Structural diagram for a combination of a two-speed electric motor with a non-uniform group:
(a) of six transmissions; (b) of four transmissions

Since the number of speed steps in the drive is $z = p_1 p_2 p_3 \dots p_m$,

$$z = x_m p_m \text{ and } x_m = \frac{z}{p_m} \tag{69}$$

The range ratio of the last extension group is

$$R_m = \varphi^{x_m(p_m-1)} = \varphi^{z - \frac{z}{p_m}} \tag{70}$$

From this it is evident that the range ratio of the last extension group will be minimum for the minimum value of $p_m = 2$. Thus

$$R_m = \varphi^{z - \frac{z}{2}} = \varphi^{\frac{z}{2}} \tag{71}$$

To avoid the necessity of introducing a multiplier device, the range ratio R_m should not be more than R_{lim} which is conditioned by the limiting transmission ratios:

$$R_m \leqslant R_{lim} = \frac{i_{max\ lim}}{i_{min\ lim}} = 8 \text{ to } 10 = C$$

At a value $p_m = 2$, it follows from equation (71) that $\varphi^{\frac{z}{2}} \leqslant C$, hence

$$\varphi^{z-1} \leqslant \frac{C^2}{\varphi}$$

or

$$R_{dr} = \varphi^{z-1} \leqslant \frac{C^2}{\varphi} \tag{72}$$

At a progression ratio $\varphi = 1.26$, the limiting value of the range ratio of the drive is

$$R_{dr} = \frac{8^2}{\varphi} = \frac{64}{1.26} \cong 50$$

Any further increase in R_{dr} will complicate the spindle drive by the introduction of a multiplier device in the last extension group.

The use of carbide-tipped cutting tools along with high-speed steel tools in general-purpose machine tools required a 2- to 4-fold increase of the range ratio of the drive above the limiting value $R_{dr} = \frac{64}{\varphi}$.

Under these conditions, the normal uniform drive structure is rejected to avoid the introduction of a multiplier device which leads expediently to a drive of combined structure with $z = z_0 (1 + 1)$ or $z = z_0 (1 + 2)$ as, for example, in the multiplier device of a divided drive (see pp. 58 and 79). In such cases, the range ratio of the last extension group can be reduced by

employing: (1) overlapping (repetition) of a part of the spindle speed steps, (2) drives with a broken geometrical series, and (3) drives with a combined structure.

Overlapping speed steps. Two methods may be used to increase the range ratio of a drive above the limiting value $R_{dr} = \frac{64}{\varphi}$ by overlapping a part of the speed steps.

The first method consists in reducing the characteristic of the last extension group by several units in comparison with the calculated value. At a number of transmissions $p_m = 2$ in the last extension group (more advantageous than when $p_m = 3$), with a range ratio in each of the groups of the drive not exceeding $R_{lim} = 8$, and with the total number of transmissions in the groups being only one more than the minimum number of transmissions for a normal uniform structure, a maximum total range ratio of the drive equal to $R_{max} \cong 400$ is obtained for $\varphi = 1.26$ and for the maximum number of speed steps $z = 27$ (Fig. 22*a*). Likewise, $R_{max} \cong 360$ is obtained for $\varphi = 1.41$ and a number of speed steps $z = 18$ (Fig. 22*b*).

In the second method, the speed steps are overlapped by a shift of the series of speed steps when engagements are made in the transmissions of the shift group. To obtain a sufficiently large number of speed steps (up to $z = 24$ at $\varphi = 1.26$), the total number of transmissions in the groups are increased by four or five transmissions in comparison with the normal structure.

Broken geometrical series. Academician A. Gadolin proposed the geometrical series of spindle speeds for machine tools (see p. 29) on the basis of the equal probability of operation at all spindle speed steps within the whole range of variation. To adopt the spindle drive mainly for the machining of medium-size work (in reference to the capacity of the given machine tool), and taking into consideration the possibility of handing over work, near to the limiting sizes (maximum and minimum), for machining in machine tools of adjacent sizes in the same size range, a broken geometrical series is employed with a progression ratio φ_1 for the middle speeds and with $\varphi_2 > \varphi_1^2$ for the extreme speed steps in the range of speed variation. This reduces the number of speed steps and the number of transmissions (in comparison with a normal uniform structure), simplifies the construction and makes it possible to increase the range ratio of the spindle drive without changing the limiting transmission ratios and without introducing a multiplier device.

Following the whole complex of transmissions which provide z_k speed steps with the progression ratio φ_2, we engage a multiplier group of two transmissions having a characteristic x', referred to the progression ratio φ_2, of such a value that a symmetrical broken series is obtained (Fig. 23) which has $(u + 1)$ middle terms with the progression ratio $\varphi_1 = \sqrt{\varphi_2}$.

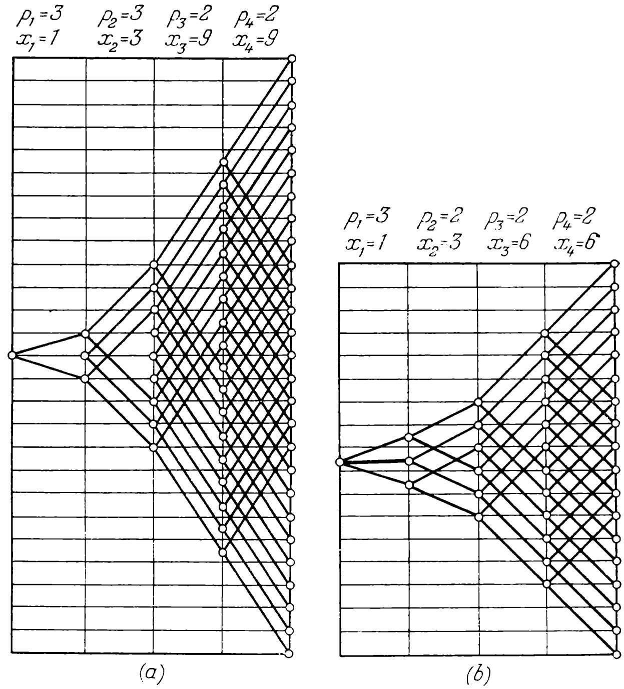


Fig. 22. Structural diagram of a drive with overlapping speed steps: (a) at $\varphi = 1.26$; (b) at $\varphi = 1.41$

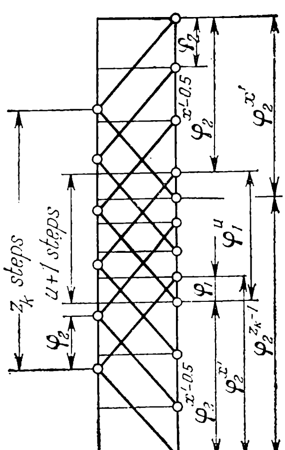


Fig. 23. Constucting the zone of a structural diagram for a multiplier transmission group providing a broken geometrical series of speeds

The range ratio of the drive (see Fig. 23) is

$$R_n = \varphi_2^{z_k - 1}\varphi_2^{x'} = \varphi_2^{(x' - 0.5)2}\varphi_1^{u} \tag{73}$$

from which

$$(z_k - 1) + x' = (x' - 0.5)2 + \frac{u}{2}$$

and

$$x' = z_k - \frac{u}{2} \tag{74}$$

The number of speed steps in the drive is $z = 2z_k$, since two transmissions have been accepted for the multiplier group. Hence, $z_k = \frac{z}{2}$ and the last equation can be written as

$$x' = \frac{z - u}{2} \tag{75}$$

The characteristic of the multiplier group can be found from these equations.

Maximum range ratio of a drive, taking the limiting transmission ratios into consideration. When two transmissions are provided in the last extension group, the maximum range ratio in the whole complex of z_k speed steps with a progression ratio φ_2 is limited by the value [see equation (72)]

$$R_k = \varphi_2^{z_k - 1} \leqslant \frac{C^2}{\varphi_2}$$

Assuming that $C = \varphi_2^r$, then $\varphi_2^{z_k} \leqslant \varphi_2^{2r}$ and

$$z_k \leqslant 2r \tag{76}$$

The limiting range ratio for multiplier groups consisting of two transmissions is

$$R_{mul} = \varphi_2^{x'} \leqslant C = \varphi_2^r$$

from which

$$x' \leqslant r$$

Spindle drives with a combined structure. A combined structure constitutes the sum of the structures of two drives, one designed for the higher and the other for the lower speed steps.

The structural formula of a combined drive is of the form

$$z = z_0(z' + z'')$$

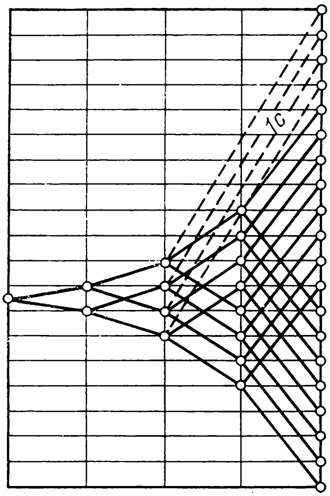


Fig. 24. Structural diagram for a drive with the structural formula $z = z_0\,(1+z'')$

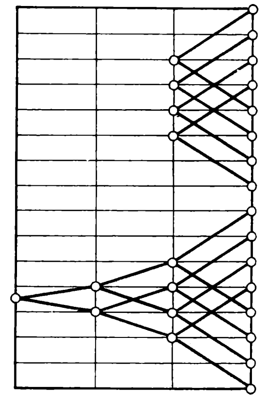


Fig. 25. Structural diagram for a drive with the structural formula $z=z_0(2+2)$

where z_0 = number of speed steps in the part common to the component drives

z' = number of speed steps in the high-speed component of the drive

z'' = number of speed steps in the low-speed component of the drive.

In respect to the total number of transmissions in the groups of combined drives, constructed in accordance with structural formulas $z = z_0\,(1 + z'')$ and $z = z_0\,(2 + 2)$ (Figs. 24 and 25, respectively), these drives can be as advantageous as drives with a normal multiplier structure having the same number of speed steps.

The maximum range ratios without introducing a multiplier device, for a drive with $z = z_0\,(1 + z'') = z_0 + z_0 z''$ and having $z_0 z''$ speed steps in the main part of the drive and z_0 steps in the supplementary part, is

$$R_{max} = R_{main}\varphi R_{sup} = R_{main}\varphi\varphi^{z_0-1} \leqslant \frac{C^2}{\varphi}\,\varphi^{z_0} \tag{77}$$

For example, at $\varphi = 1.26$ and $z_0 = 6$,

$$R_{max} \leqslant \frac{C^2}{\varphi}\,1.26^6 = 4\,\frac{C^2}{\varphi}$$

which is fourfold that of a normal multiplier group.

Examples of the application of combined drive structure are the divided spindle drive of the Soviet engine lathe, model 1616, and the spindle drives of engine lathes, models 1A62 and 1K62.

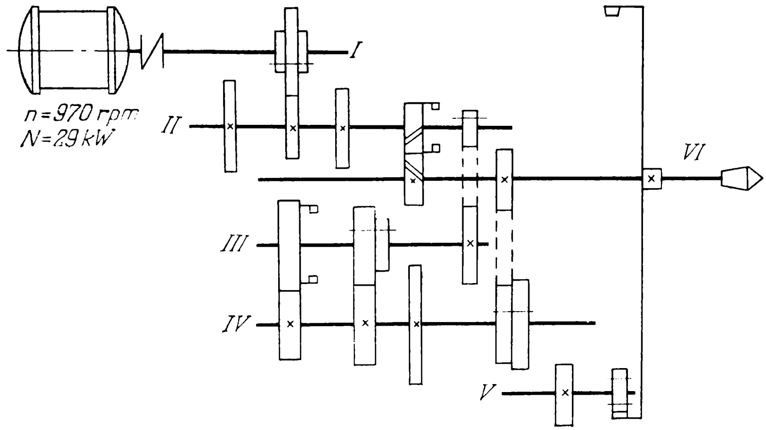


Fig. 26. Kinematic diagram of the spindle drive of a heavy lathe

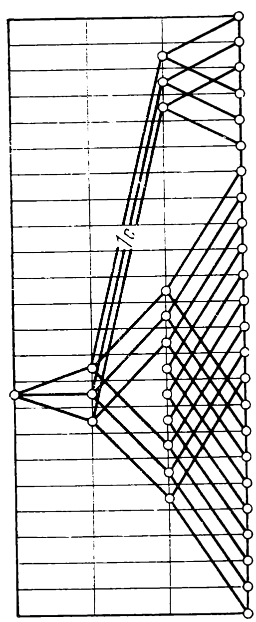


Fig. 27. Structural diagram of the drive shown in Fig. 26

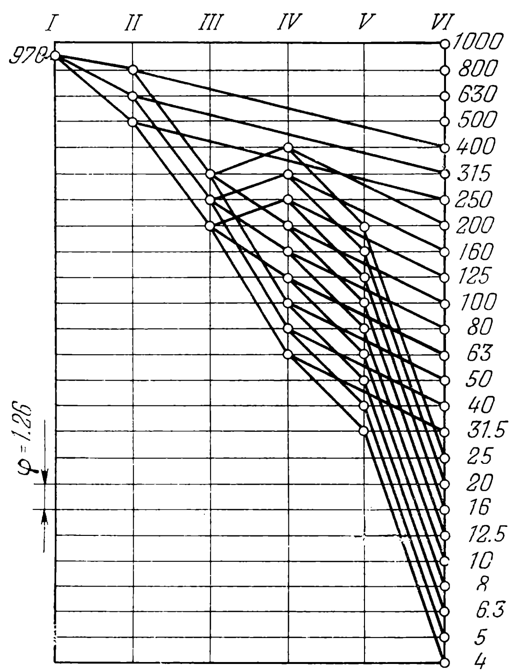


Fig. 28. Speed chart of the drive shown in Fig. 26

Figures 26, 27 and 28 show the gearing diagram, structural diagram and speed chart of a heavy lathe in which the spindle drive is of combined structure.

Advantages of a combined-structure drive are: the range ratio is extended without the introduction of a multiplier device; the gear train is shortened at the high-speed steps, as well as at the medium-speed steps in heavy machine tools (see Figs. 27 and 28), thereby reducing friction losses and facilitating speed-up and braking of the drive; and the possibility of using different types of transmission to the spindle for the high- and low-speed steps, such as belt and toothed gearing transmissions for a divided drive, or helical and internal gearing in the spindle drives of heavy machine tools (see Fig. 26).

3-3. Determining the Number of Teeth on the Gears of Group Transmissions

All the Gears in the Group Have the Same Module

If the centre-to-centre distance is maintained constant and all the gears of a group have the same module, then

$$z_j + z_j' = S_z = \text{const} \tag{78}$$

where z_j and z_j' = numbers of teeth, respectively, of the driving and driven gears of a pair, and $j = 1, 2, 3, \ldots, p$

S_z = sum of the numbers of teeth of the meshing gears.

$$i_j = \frac{z_j}{z_j'} \tag{79}$$

Combining equations (78) and (79),

$$\left.\begin{aligned} z_j &= \frac{i_j}{i_j+1} S_z \\ z_j' &= \frac{1}{i_j+1} S_z \end{aligned}\right\} \tag{80}$$

Using these last two formulas, the numbers of teeth of all the gears in the group are found from their given sum S_z.

The value S_z is usually determined by the method of the least common multiple.

If $i_j = \frac{z_j}{z_j'} = \frac{a_j}{b_j}$, where a_j and b_j are mutually prime whole numbers,

then equations (80) can be written as

$$\left.\begin{aligned} z_j &= \frac{a_j S_z}{a_j + b_j} \\ z'_j &= \frac{b_j S_z}{a_j + b_j} \end{aligned}\right\} \tag{81}$$

Hence, if z_j and z'_j are to be whole numbers, it is necessary that S_z be a multiple of the sum $a_j + b_j$.

In a group consisting of p gear transmissions with transmission ratios $i_j = \frac{a_j}{b_j}$, where $j = 1, 2, 3, \ldots, p$, the minimum sum $S_{z\,min}$ of the number of teeth of any pair of meshing gears is equal to K, the least common multiple of the sums

$$a_1 + b_1,\ a_2 + b_2,\ a_3 + b_3,\ \ldots,\ a_p + b_p \tag{82}$$

$$S_{z\,min} = K$$

If, for the value $S_{z\,min}$ thus found, the number of teeth of the pinion (smaller gear) in the transmission with $i = i_1$, i.e., for a number of teeth $z_1 = \frac{a_1}{a_1 + b_1} S_z$ (using the notation given above), is inadmissibly small, the number of teeth of this gear is increased by a whole number E times so that an acceptable value is obtained. Thus

$$z_1 = z_{min} E = \frac{a_1}{a_1 + b_1} E S_{z\,min} \tag{83}$$

accordingly, the sum of the numbers of teeth is

$$S_z = E S_{z\,min} = EK \tag{84}$$

Another factor that must be taken into consideration in solving such problems is the maximum permissible peripheral speed of the larger gear which, at a given maximum rotational speed, is proportional to the pitch diameter of the gear and, consequently, to its number of teeth.

Example. Determine the numbers of teeth of the gears in the transmissions of a group specified by the zone of a speed chart shown in Fig. 29. It is evident from the chart that

$$\left.\begin{array}{l|l} i_1 = \dfrac{1}{\varphi^3} = \dfrac{1}{1.26^3} = \dfrac{1}{2} & 1 + 2 = 3 \\ i_2 = \dfrac{1}{\varphi^2} = \dfrac{1}{1.26^2} = \dfrac{1}{1.58} \cong \dfrac{7}{11} & 7 + 11 = 18 \\ i_3 = \dfrac{1}{\varphi} = \dfrac{1}{1.26} \cong \dfrac{4}{5} & 4 + 5 = 9 \\ i_4 = \dfrac{1}{\varphi^0} = \dfrac{1}{1.26^0} = \dfrac{1}{1} & 1 + 1 = 2 \end{array}\right\} \begin{array}{l}\text{Least} \\ \text{common} \\ \text{multiple} \\ K = 3 \times 3 \times 2 = 18\end{array}$$

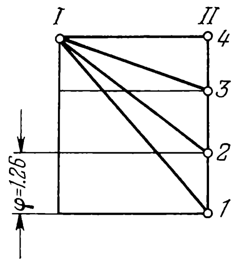


Fig. 29. Zone of a speed chart for a group of four transmissions

Then, for $S_{z\,min} = K = 18$, we obtain $z_{min} = z_1 = = \frac{1}{1+2} 18 = 6$.

Let us assume that in the speed gearbox being designed it has been specified that no gear shall have less than 22 teeth. This being the case, $E \geqslant \frac{22}{6} > 3$, and it is necessary to take $E = 4$. Hence

$$S_z = ES_{z\,min} = 4 \times 18 = 72$$

Having established this sum of the numbers of teeth, there should be no difficulty in finding the numbers of teeth of all the gears by the formulas (81).

The value $S_z = 72$ is of interest in that with this sum of the numbers of teeth of each pair of meshing gears and with transmission ratios ranging from $i_{min} = \frac{1}{2}$ to $i_{max} = \frac{2}{1}$, the arithmetical series of the numbers of teeth on the driving gears and, consequently, on the driven gears with a difference $\delta = 1$, coincide, with an accuracy sufficient for practical purposes, with the geometrical series of transmission ratios having a progression ratio $\varphi = 1.06$. Therefore, using this value of S_z we can obtain any standard transmission ratio $i = 1.06^{\pm E}$ within the above-mentioned limits of i.

The Gears in the Group Have Different Modules

In this case, the number of teeth of the driving gear can be expressed in terms of the centre-to-centre distance A as follows

$$z_j = \frac{2A}{m_j} \frac{a_j}{a_j + b_j} \tag{85}$$

where m_j is the end module (in the plane of rotation) of gear z_j, and a_j and b_j have the meaning indicated on p. 60.

The number z_j will be a whole number only under the condition that $2A = = Em_j (a_j + b_j)$, in which E is the symbol of a whole number as before. Hence, the minimum doubled centre-to-centre distance $2A$ is equal to the least common multiple of the product $m_j (a_j + b_j)$ for all the transmissions in the group. If the distance turns out to be too large, the least common multiple of the modules is found and, after multiplying it by a certain whole number, is taken as the doubled centre-to-centre distance (in mm). Then, the sums of the numbers of teeth $S_z = \frac{2A}{m_j}$ are whole numbers, but the calculated values $z_j = \frac{2A}{m_j} \frac{a_j}{a_j + b_j}$ are fractional and must be rounded off.

Determining the numbers of teeth of helical gearing. The normal module m_n is the same for all the helical gears of a group; the helix angles β_j of the

gears in certain transmissions of the group may differ. The centre-to-centre distance is

$$A = \frac{m_n (z_j + z'_j)}{2 \cos \beta_j} \tag{86}$$

The numbers of teeth of meshing gears are

$$z_j = \frac{2A}{m_n} \frac{a_j \cos \beta_j}{a_j + b_j} \quad \text{and} \quad z'_j = \frac{2A}{m_n} \frac{b_j \cos \beta_j}{a_j + b_j} \tag{87}$$

There can be two principal cases.

1. If it has been specified that β_j = const = β (in case the helical gears are to be cut in a gear shaper which has only one helical guide, or it is undesirable to change the guide), the sum S_z of the numbers of teeth on the meshing pairs of gears is found as a value which is a multiple of the sums $a_j + b_j$. Then S_z is used to calculate the centre-to-centre distance and numbers of teeth of the gears by the equations

$$A = \frac{m_n S_z}{2 \cos \beta}; \; z_j = S_z \frac{a_j}{a_j + b_j} \quad \text{and} \quad z'_j = S_z \frac{b_j}{a_j + b_j} \tag{88}$$

2. If the centre-to-centre distance A is given, then K, the least common multiple of the sums $a_j + b_j$, is determined. A corresponding sum of teeth $S_{zj} = KE_j$ is taken for each transmission of the group, the whole numbers E_j being selected in such a manner that in calculating the required angles β_j, the value $\cos \beta_j = \frac{m_n KE_j}{2A}$ does not appreciably deviate from unity and the helix angles are not excessively large. The numbers of teeth of the gears are

$$z_j = \frac{KE_j a_j}{a_j + b_j} \quad \text{and} \quad z'_j = \frac{KE_j b_j}{a_j + b_j} \tag{89}$$

and now are evidently whole numbers.

3-4. Recommendations for Developing the Kinematic Scheme of a Machine Tool

Versions of Kinematic Scheme Structure

Versions of the structure of a kinematic scheme having a geometrical series of speeds are associated with the various solutions of the two principal general questions concerning the kinematics of the drive: (a) setting up (changing) the speeds and feeds within the limits of the specified range; and (b) reducing or increasing the speed of the initial (input) shaft of the drive to the specified minimum (or maximum) speed of the last working shaft of the drive.

Various structures of speed-changing facilities conform to various types of structural formulas and various versions of structural diagrams.

For a specified or selected number of transmission groups m and a specified number of transmissions in each group, there will be numerous design versions of the structural formula and structural diagrams, which differ in the actual order of arrangement of the groups along the train of the drive. Thus

$$z = p_a p_b p_c \ldots p_r = p_b p_a p_c \ldots p_r = p_c p_a p_b \ldots p_r = p_c p_b p_a \ldots p_r = \ldots \quad (90)$$

The quantity of such design versions is equal to the number of permutations of m groups, i.e., $m! = 1 \times 2 \times 3 \ldots \times m$.

If there are q groups with an equal number of transmissions in each, the number of design versions will be $\frac{m!}{q!}$.

Each of the subindexes a, b, c, . . ., r, denoting the group number in equation (90) can have a value ranging from 1 to m. Each group may be the main group, first, second or any extension group up to the very last one. Hence, for each design version of speed-changing structure, there may be $m!$ kinematic versions. Therefore, the total number of versions may be

$$\frac{m!}{q!} m! = \frac{(m!)^2}{q!} \quad (91)$$

For example, for the structure $z = 12 = 3 \times 2 \times 2$, in which $m = 3$ and $q = 2$, the number of design versions is

$$\frac{m!}{q!} = \frac{1 \times 2 \times 3}{1 \times 2} = 3$$

Each of these versions may be made up of $3! = 6$ kinematic versions. Hence, the total number of versions is

$$\frac{(3!)^2}{2!} = \frac{36}{2} = 18$$

Minimum Number of Transmissions

The total number of transmissions in the groups $S_p = p_a + p_b + p_c + \ldots + p_r$ required to obtain a specified number of speed steps $z = p_a p_b p_c \ldots p_r$ will be minimum if

$$p_a = p_b = p_c = \ldots = p_r = \sqrt[m]{z} = p \quad (92)$$

It can be shown that if m, the number of transmission groups, is not specified then the minimum number of transmissions can be obtained under the condition that

$$p = 2 \text{ or } p = 3 \quad (93)$$

Thus, it proves expedient to have $p = 2$ or $p = 3$ transmissions in each group and, since $2 + 2 = 2 \times 2 = 4$, $p = 4$ as well.

These are actually the numbers of transmission that are employed for gearing with sliding cluster gears, when the number of gears is twice the number of transmissions. This condition does not hold true for change gears, where the same pair of gears can be interchanged. The application of a broken geometrical series considerably reduces the number of transmissions required.

Minimum Number of Transmission Groups

The minimum number of transmission groups, required to obtain the specified range ratio $R_n = R_a R_b R_c \ldots R_r$, can be had in the case when

$$R_a = R_b = R_c = \ldots = R_r = R_{lim} = \frac{i_{max\,lim}}{i_{min\,lim}} \tag{94}$$

Since for any group transmission $\varphi_p = R_k \varphi$ and the range ratio $R_p = \varphi_p^{p-1} = (R_k \varphi)^{p-1}$, equation (94) can be satisfied if $\varphi = 1$, $R_k = R_1 = R_{lim}$ and $p = p_2 = 2$, i.e., if the drive consists of a group with infinite speed variation ($\varphi \to 1$), having a range ratio R_{lim}, and an extension group of two transmissions. Such a combination provides the simplest speed-changing structure.

Thus, a variable-speed drive with a range ratio $R_1 = R_{lim}$, in conjunction with an extension ratio group comprising two transmissions, provides a total range ratio $R_n = R_1 R_2 = R_{lim}^2$. For instance, at $R_{lim} = 8$, $R_n = 64$ (see p. 54).

Next in order as to its possibilities for simplifying speed-changing structure and for providing the maximum range ratio without resorting to multiplier devices is a combination of an extension group of two transmissions and a group with change gears. The number of transmissions in the latter group is not limited by axial overall dimensions, but only by the limiting transmission ratios (see p. 44).

If in this case $p_2 = 2$ and $R_2 = R_{lim} = 8$, then

$$R_1 = \frac{R_{lim}}{\varphi} = \frac{8}{\varphi} \quad \text{and} \quad R_n = R_1 R_2 = \frac{64}{\varphi}$$

Drives in which speeds are changed by uniform groups having $p = 2$ or 3 transmissions in each group provide the minimum total number of transmissions in all the groups but, at the same time, require the maximum number of groups. Thus, to reduce the number of gears it becomes necessary to increase the number of shafts, bearings and bores in the gearbox housing required to make up the group transmissions. In many cases, these elements of group

transmissions are used to set up the reduction gear train. Therefore, in comparing the different versions of speed-changing arrangements on the basis of the amount of transmissions, gears, shafts and other design elements required, it is also necessary to take into account the gear train required purely for speed reduction.

In the case of long reduction trains (heavy machine tools), it proves expedient to change speeds by means of uniform groups having the minimum number of transmissions in all the groups and $p = 2$, 3 or 4 transmissions in each group. More advantageous for short reduction trains (in small high-speed machine tools) is a simplified speed-changing structure with a minimum number of groups and with the use of nonuniform groups (see Figs. 19, 20 and 21).

Taking the Weight of the Drive into Account

The size of the elements in the transmissions of a drive increases with an increase in the torque developed in the shafts of the drive.

The torque is equal to

$$M_t = 71{,}620 \frac{N}{n} \eta \text{ kgf-cm} \tag{95}$$

where N is the power in kW. Thus

$$\log n + \log M_t = \log 71{,}620 + \log N + \log \eta \tag{96}$$

To represent these relationships on a logarithmic graph of the drive speeds (Fig. 30), at a distance of $\log 71{,}620 + \log N$ from the horizontal line $\log 1 = 0$, we construct the line AB which is the power transmitted from shaft I to shaft V without loss. If the efficiency is the same for the transmissions between the shafts, the inclined line AC represents the power transmitted if the transmission losses are taken into consideration.

The intercept of the shaft line between line AC and any point indicating the speed of this shaft represents in a logarithmic scale the magnitude of the torque transmitted by this shaft at the given speed, such as those shown for the torques loading shaft III.

In plotting the reduction transmission trains from n^{I} to n^{V} (Fig. 31) for the given transmitted power (line AC), the design torques developed in the first and last shafts have quite definite magnitudes.

The design torques developed by the intermediate shafts will depend upon the structure of the reduction train. If it is constructed along line aqe (Fig. 31), using a transmission with $i_{min\ lim}$ at the beginning of the train and with $i_{max\ lim}$ at the end, the maximum torques will be obtained for the intermediate shafts. Such a design transmission train is the least advantageous.

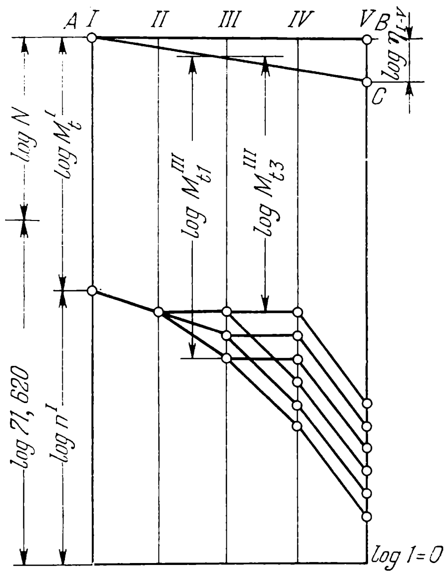


Fig. 30. Integrated diagram of speeds, transmitted power and torques

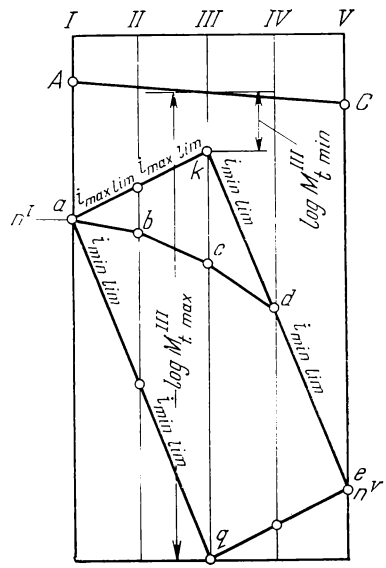


Fig. 31. Torques for various versions of the speed reduction train

If, on the other hand, the reverse arrangement of transmission ratios is used, as in line *ake*, the design torques developed in the intermediate shafts will be the minimum possible, and the weight of the drive may be at a minimum. In this case, however, group transmissions cannot be used for the first two links. Therefore, it is sound practice to design a reduction train of the type shown by broken line *abcde*, in which the transmission ratios are reduced to a greater and greater degree as the train approaches the spindle, and with the transmission ratio at the spindle taken equal to the limiting minimum value. To reduce the weight of the drive, it is expedient to apply a structural version in which the number of transmissions in the groups decreases along the train from the electric motor to the spindle. Thus, if

$$z = p_a p_b p_c \ldots p_r$$

it is advisable to make

$$p_a > p_b > p_c > \ldots > p_r$$

Simple transmissions should be arranged nearer to the spindle.

For a given total number of transmissions, this design version of drive structure ensures a larger number of transmissions whose components are

of less weight, and less transmissions with heavier components, since the design torque of the shafts increases along the train of transmission from the electric motor to the spindle.

General Considerations Influencing the Selection of the Kinematic Version of Drive Structure

In the general case, the most advantageous of all the possible kinematic versions of drive structure is one in which the characteristic of the groups increases from the electric motor to the spindle, i.e., the number of the group increases in the kinematic order of arrangement.

Thus, if

$$z = p_a p_b p_c \ldots p_r$$

then

$$a < b < c < \ldots < r$$

or, otherwise,

$$z = p_1 p_2 p_3 \ldots p_m \text{ and } x_1 < x_2 < x_3 < \ldots < x_m \qquad (97)$$

The advantage of such a sequence of group characteristics is that, for the same minimum speeds of the intermediate shafts, their maximum speeds are less. This enables the components of the transmissions to be manufactured to lower accuracy requirements, reduces the dynamic loads in the transmissions, reduces the danger of vibration, reduces wear on the components, reduces the part of the friction losses that do not depend upon the load, and increases the efficiency at high spindle speeds.

Conditions concerning overall dimensions. *Reducing the radial dimensions.* The following condition expresses the kinematic means for reducing the radial dimensions of group drives:

$$i_{min} i_{max} = 1$$

This condition leads to a symmetrical arrangement of the rays of a given group in the speed chart of the drive.

Another method of reducing radial dimensions is to make the axes of the shafts of adjacent transmission groups coincident.

Reducing the axial dimensions. As a rule, it becomes necessary to reduce the axial dimensions in cases when the spindle head is to be traversed along ways perpendicular to the spindle axis (in radial drills, planer-type milling machines and horizontal boring machines). To increase the stability of the head on its ways and to avoid vibration, it is desirable to locate rapidly rotating parts within the zone of the ways.

Measures reducing axial dimensions are: (a) arrangement of the simple transmissions among the group transmissions, (b) linked gears, i.e., gears serving as driven members in one transmission group and as driving members in the next group. The use of linked gears reduces the total number of gears in two adjacent groups.

The use of a single linked gear in a group does not introduce any kinematic restrictions; the use of two linked gears in the same group limits the total transmission ratio of the two adjacent groups. If three linked gears are used in a single group, it is impossible to obtain a geometrical series of speeds for the whole complex of speed steps (without overlapping).

Reducing friction losses in the drive. The size of the elements of a drive is determined on the basis of the transmission train engagements that give the lowest spindle speed n of all the speed steps that utilize the full power developed by the electric motor. At higher speeds n, the elements of the drive are capable of transmitting more power than is available from the motor and are therefore underloaded. This underload increases with an increase in speed n, as does the part of the friction losses that does not depend upon the load. This leads to a reduction in efficiency at high speed steps and is especially marked in general-purpose machine tools with a wide range of spindle speed variation.

These losses can be decreased by shortening the transmission train for high spindle speeds n, arranging the kinematic scheme so that a part of the transmissions, not required to obtain these speeds n, are cut out when they are engaged.

The use of a shorter transmission train, from which superfluous links have been excluded, is conducive to better dynamic conditions in speeding up and braking the drive. This may be of vital importance in general-purpose machine tools having a wide range of spindle speeds.

The principal method used to shorten the transmission train for the higher steps of spindle speed is the application of a combined structure in the drive. This feature has found fairly wide use in small and medium-size machine tools in the form of a divided drive with a belt transmission to the spindle head which contains counter gearing. The structural formula for such a drive is

$$z = z_0(1 + z'') = z_0(1 + 1) \text{ or } z = z_0(1 + 2)$$

At high speed steps, the counter gearing is cut out and the spindle is driven by the belt transmission.

An example of a geared headstock with a combined structure for a heavy lathe is shown in Fig. 26.

The spindle drive of the model 1A62 engine lathe (see Fig. 35) has the structural formula $z = z_0(1 + z'') = 2 \times 3(1 + 2 \times 2 - 1) - 1 = 23$,

since one speed step is overlapped in the last extension group and one in the combined structure.

Taking the purpose of the machine tool into account. The relative importance of the various considerations, that must be taken into account in working out the kinematic scheme of a new machine tool, is determined by the purpose of the machine tool. From general considerations (see p. 68), in a drive with the structure $z = p_1p_2p_3 \ldots p_m$, it is of advantage to take $p_1 > p_2 > p_3 > \ldots > p_m$; $x_1 < x_2 < x_3 < \ldots < x_m$ and $i_{1\,min} > i_{2\,min} > i_{3\,min} > \ldots > i_m i_{min\,lim}$. Deviations from the optimum structural version of the drive may be due to features associated with the purpose of the machine tool. Hence, the taking account of these deviations is a good method for revealing the specific features of the drives in various types of machine tools when comparing their constructions. Spindle drives are representative in this respect.

Typical of machine tools for roughing work of large diameter or those employing cutting tools of large diameter is a transmission with internal gearing, arranged outside of the spindle head or headstock. The weight of the drive is reduced by making the ratio of the internal gearing as small as possible, from about $\frac{1}{6}$ to $\frac{1}{8}$.

Machine tools for roughing with high torques at high spindle speeds (medium work diameters in multiple-tool lathes, roughing with carbide-tipped tools, etc.) frequently have helical gearing with a ratio of $\frac{1}{3}$ to $\frac{1}{4}$, operating with high design stresses. This reduces the peripheral speed of the gears and promotes smoother operation.

Machine tools for high-speed finishing operations, requiring smooth spindle rotation, may be designed with built-in electric motors, belt drives with the spindle relieved of the belt tension, pneumatic motors, as well as other drives which follow in the order of decreasing spindle speeds: belt drives in which the spindle is not relieved of belt tension; drives with a plastic helical gear meshing with a cast iron or steel gear having tooth surfaces with a hardness of at least $40R_C$; speed-up helical gearing drives; spiral bevel gearing drives; and reduction spur or helical gearing drives.

The limits within which the machining conditions (speeds and feeds) are varied may have an influence on the type of spindle drive used. If there is to be no change in the nature of the machining conditions, a drive complying with these conditions is applied. If the conditions are to be changed within comparatively narrow limits, a single transmission, adapted to this range ratio, is used for the spindle drive. For example, in the model 1Д62 engine lathe, this is a transmission through helical gears having a ratio of 1 : 2 to avoid high driving shaft speeds and high peripheral speeds of the spindle gear.

Two transmissions to the spindle are used if the nature of the machining conditions is to vary in wide limits. Examples are: a speed-up and a reduction transmission with spur or helical gears; reduction gear transmission and a belt transmission with the spindle relieved of the belt tension; and an internal and external spur or helical gear transmission.

In some cases the last extension group has certain specific features.

In engine lathes, the range ratio R_m of this group is used, not only for extending the range ratio of the spindle speeds, but also for increasing the pitch of threads that can be cut. This range ratio should be $R_m = 2^E$, where E is a whole number. For this same purpose the extension group is arranged near the spindle.

One desirable feature of turret lathes is the possibility of making rapid engagements in the last extension group without stopping the lathe by means of friction clutches. Consequently, this group should not be at the end of the transmission train where large torques occur.

In a similar manner, the system for changing speeds and the structure of other group transmissions depend upon the purpose of the machine tool.

In cases when the machine time is small, the handling time required to change speeds should also be small. Speeds should be changed without stopping the machine tool or a system of speed preselection should be used. The system applied to change the speeds affects the structure of the drive and should be taken into account in working out the kinematic scheme.

CHAPTER 4

SPEED AND FEED GEARBOXES. STEPLESS DRIVES

4-1. Speed Gearboxes in Machine Tools

General Principles. Requirements Made to Speed Gearboxes

The construction of a speed gearbox is intimately linked with the whole structure of the spindle drive (see Chap. 3).

The speed gearbox can be built into the spindle head housing in which case it is called the spindle head or, on the contrary, the spindle head is called the speed gearbox. If the speed gearbox is arranged in a separate housing and linked to the spindle head through some type of transmission, it is usually called either a reducing gear or a speed gearbox regardless of whether the last extension group has been housed in the spindle head or not.

The scheme for the speed gearbox is worked out together with that of the whole spindle drive. Here, the general requirements, listed above (p. 68) in respect to the kinematic scheme, should be taken into consideration in accordance with their relative importance, which is determined by the purpose of the machine tool. For example, smoothness of spindle rotation is of more importance to a finishing machine than to one used for roughing operations.

The speed gearbox should provide the designed series of spindle speeds from n_{min} to n_{max} according to the standard H11-1 (in the USSR), the deviations being within the permissible values stipulated by this standard (see p. 31), and transmit power of an amount dictated by the purpose of the machine tool. Smooth silent operation of the transmission and accurate vibrationless rotation of the spindle are factors necessary to obtain machined surfaces of the specified accuracy and finish. These properties of a spindle head can be ensured by sufficiently rigid housing, shafts, spindle and bearings; by an expedient arrangement of the electric motor and by manufacturing and assembling the elements of the drive to a sufficiently high degree of accuracy.

Components of the drive should be manufactured to an accuracy which increases with the maximum speeds of the intermediate shafts and the maximum peripheral speeds of the gears, especially those driving the spindle. The degree of accuracy of toothed gears should always suit their maximum peripheral speed. The selection of the class of accuracy of ball and roller bearings should always be well grounded. Since the cost of these bearings increases drastically with their class of accuracy (see p. 120), bearings of the lowest feasible class of accuracy should be applied in all cases.

The controls of the speed gearbox should be in line with the conditions indicated in Sec. 11-1.

Mechanisms of speed gearboxes should be easily accessible for observation during operation, for periodic inspection in carrying out preventive maintenance, and for making adjustments in bearings, clutches, brakes and components of the control system.

The speed gearbox housing should have proper seals or packings at points where shafts extend from the housing and in the joints of all covers to prevent oil from leaking out, and dirt, abrasive particles and cutting fluid from getting in.

Producibility requirements made to speed gearboxes are considered in detail in books on engineering manufacturing processes. The most important of these requirements are:

1. The construction should be as simple as possible. This is characterized to a considerable degree by the total number of shafts, gears, clutches, bearings and control system components.

2. All the parts should permit convenient machining. This especially concerns the housing since it requires the highest labour input. The number of holes whose axes do not coincide should be as few as possible. The diameters of holes on a single axis should decrease in one direction along the axis so that the holes can be bored without turning the housing 180°. This requirement need not be complied with if the housing is to be machined in a two-way unit-built boring machine in large-lot production or on the revolving table of a horizontal boring machine in piece or small-lot production. It is good practice to arrange pads and lugs in a single plane on the external surfaces of the housing to permit several housings to be milled or planed simultaneously in one line. Internal faces and thread in bored holes may lead to complications in machining and should be avoided wherever possible.

3. The labour input in speed gearbox manufacture can be substantially reduced by reducing the number of new parts, replacing them by standard parts; limiting the number of different fits and different gear modules and by simplifying the configuration of large gears.

4. For the same reasons it is desirable to unify the construction of the gearbox with that of earlier models and to use subassemblies whose production has been mastered by the given plant.

5. Assembly should be as simple as possible and have a minimum amount of fitting operations. Best practice consists in separate assembly of all the subunits and of the whole gearbox which is subsequently mounted on the machine tool.

6. The construction of the joint between the housing and the bed or base should ensure convenient aligning of the speed gearbox or spindle head. There should be no clearances in the joints at the places where the clamping screws are located.

Manufacturing Specifications

The speed gearbox housing is usually made of cast iron, grade СЧ15-32, according to USSR Std GOST 1412-54, and in certain cases of cast iron of higher quality, for example grade СЧ28-48. If the housing is of complex or intricate configuration, the casting should be aged to relieve internal stresses.

In some models of precision machine tools, the spindle head and speed gearbox housings are cast of invar (36% Ni and the remainder Fe) in which the coefficient of thermal expansion is 1 to 2×10^{-6}, i.e., from $\frac{1}{5}$ to $\frac{1}{10}$ that of ordinary cast irons. This is done to exclude the effect of temperature deformations of the housing on the operating accuracy of the machine tool.

Tolerances are assigned on the centre-to-centre distances of the shafts in a train of gearing. The holes for the shaft bearings are usually bored to the second grade of accuracy. The tolerances on the form of the bore should not exceed $\frac{1}{3}$ to $\frac{1}{2}$ of the tolerance zone of the bore diameter.

Tolerances are assigned on the bores for the spindle bearings in accordance with the requirements specified for the accuracy and surface finish of workpieces that are to be machined. In most cases, spindle bearing bores are machined to the first grade of accuracy.

The out-of-squareness of surfaces of the housing, perpendicular to axes of the bores, is commonly limited to 0.01 to 0.03 mm on a radius of 100 mm. Much wider tolerances (0.2 to 0.5 mm) are assigned for the distances between these surfaces because the exact axial position of the shafts does not affect the operation of the gearbox, while the misalignment of covers, located from the end faces of the housing, may lead to one-sided contact of the covers with the outer rings of the bearings and to consequent cocking of the bearings.

The spindle axis should be parallel to the base surface of the spindle head housing. The axes of the shafts should be parallel to each other. The holes for the spindle bearings should be strictly coaxial.

In up-to-date models of machine tools all the gears of the speed gearbox are properly hardened and ground.

Design Gear Trains of the Gearbox

In the same way as the whole spindle drive, the gearbox has two design gear trains, i.e., engagements on which its design is based. They correspond to the minimum spindle speed n_{min} for operation with carbide-tipped tools

and with high-speed steel tools. The second train should be calculated for a power requirement $\frac{1}{2}$ to $\frac{1}{3}$ that of the first train.

In general-purpose machine tools, the minimum speeds of the spindle are employed for operations which do not require the full installed power of the electric motor, such as cutting thread, reaming, etc. In designing machine tools of this type, the gear trains for obtaining the first fourth of the series of spindle speeds are not calculated on the basis of the installed power of the drive motor if the machine tool is intended for the use of high-speed steel tools. Similar standards have not yet been established for machine tools intended for the use of carbide-tipped tools.

In a gearbox, the calculated power that can be transmitted by the weakest link of the train is increased with the spindle speed.

Gearbox Efficiency

The underloaded condition of a drive at the high steps of spindle speed, in comparison with the available design power, is one of the main reasons for the low efficiency of the drive at these speeds. For this reason, it is desirable to shorten the gear train at high speeds to increase the efficiency (see p. 69) in such manner as to leave the least possible transmissions that are underloaded in respect to the available design power. In effect, this involves shortening the train by cutting out the last links which are of large size.

Examples of such a solution are the engine lathe headstock with a divided drive (Fig. 32) in which the countergearing is cut out, and the extension group of the headstock (see Fig. 35) of the model 1A62 lathe.

Investigations carried out in ENIMS by G. Levit led to the proposal of the following semiempirical formula for tentatively determining the no-load power of a gearbox at various spindle speeds:

$$N_{nl} = \frac{K_m d_m}{955 \times 10^3} (n_{\mathrm{I}} + n_{\mathrm{II}} + n_{\mathrm{III}} + \ldots + C n_{sp}) \text{ kW} \tag{98}$$

where K_m = 30 to 50 = factor whose value is the lower, the better the manufacturing conditions, lubrication and construction in respect to friction losses in the elements of the drive

d_m = mean diameter of the shafts in the gear train (except for the spindle diameter), cm

$n_{\mathrm{I}}, n_{\mathrm{II}}, n_{\mathrm{III}} \ldots$ = speeds, rpm, of those shafts in the drive that are included in the gear train for the given value of n_{sp}

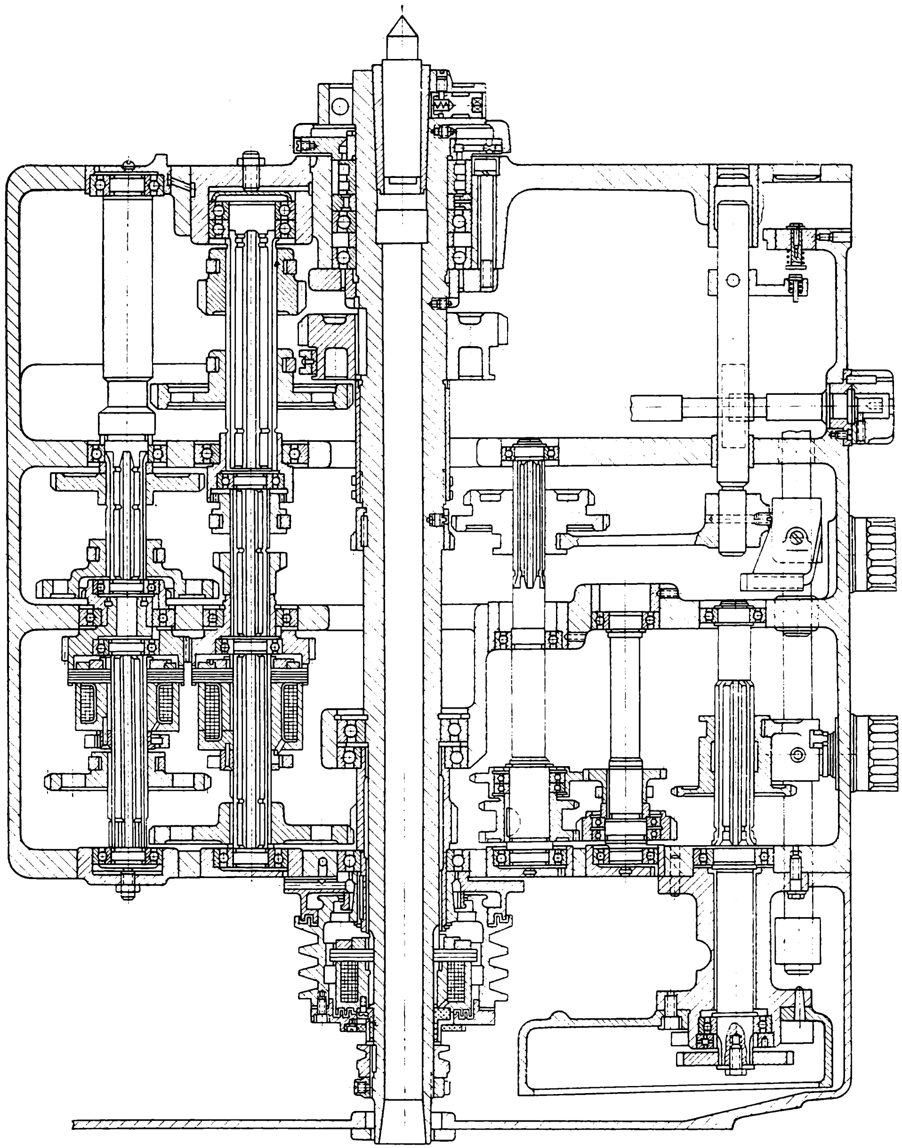


Fig. 32. Development of the headstock of the model 1620 engine lathe

n_{sp} = spindle speed at which the no-load power requirement is to be determined, rpm

$C = k_{sp}\frac{d_{sp}}{d_m}$ = coefficient; $k_{sp} = 2$ for spindles running in sleeve bearings and $k_{sp} = 1.5$ for spindles running in ball or roller bearings.

Concerning friction losses, see also page 69.

4-2. Types of Speed Gearboxes

Gearboxes are classified according to their layout and the method used for changing speeds.

Gearbox Layout

The layout of the gearbox in the machine tool being designed is affected both by the general layout of the whole machine tool, i.e., the spatial and dimensional relations between the various units, and the spatial relations between the gearbox proper and the spindle head (or headstock) containing the spindle with its supports and transmissions.

The layout of a gearbox depends to a considerable extent, as does that of the whole machine tool, upon the purpose of the machine tool and its type and size. This accounts for the great variety of construction layouts of gearboxes, notwithstanding the limited number of constructional elements of which they consist.

The gearbox layout adopted for the machine tool being designed should be thoroughly substantiated.

I. Gearboxes built into the spindle head (or headstock). Such gearboxes are employed in most medium-size and heavy machine tools. The advantages of this layout are: a more compact spindle drive, higher concentration of controls, fewer housing-type parts and less assembly work involving the fitting of jointing surfaces.

Drawbacks of this layout are the possibility of transmitting vibration from the gearbox to the spindle, heating of the spindle head by heat evolved in the gearbox, and the difficulty of employing a flexible transmission to the spindle along with toothed gearing.

The following types of dimensional layouts of spindle heads (headstocks) with built-in gearboxes find application in modern machine tool engineering:

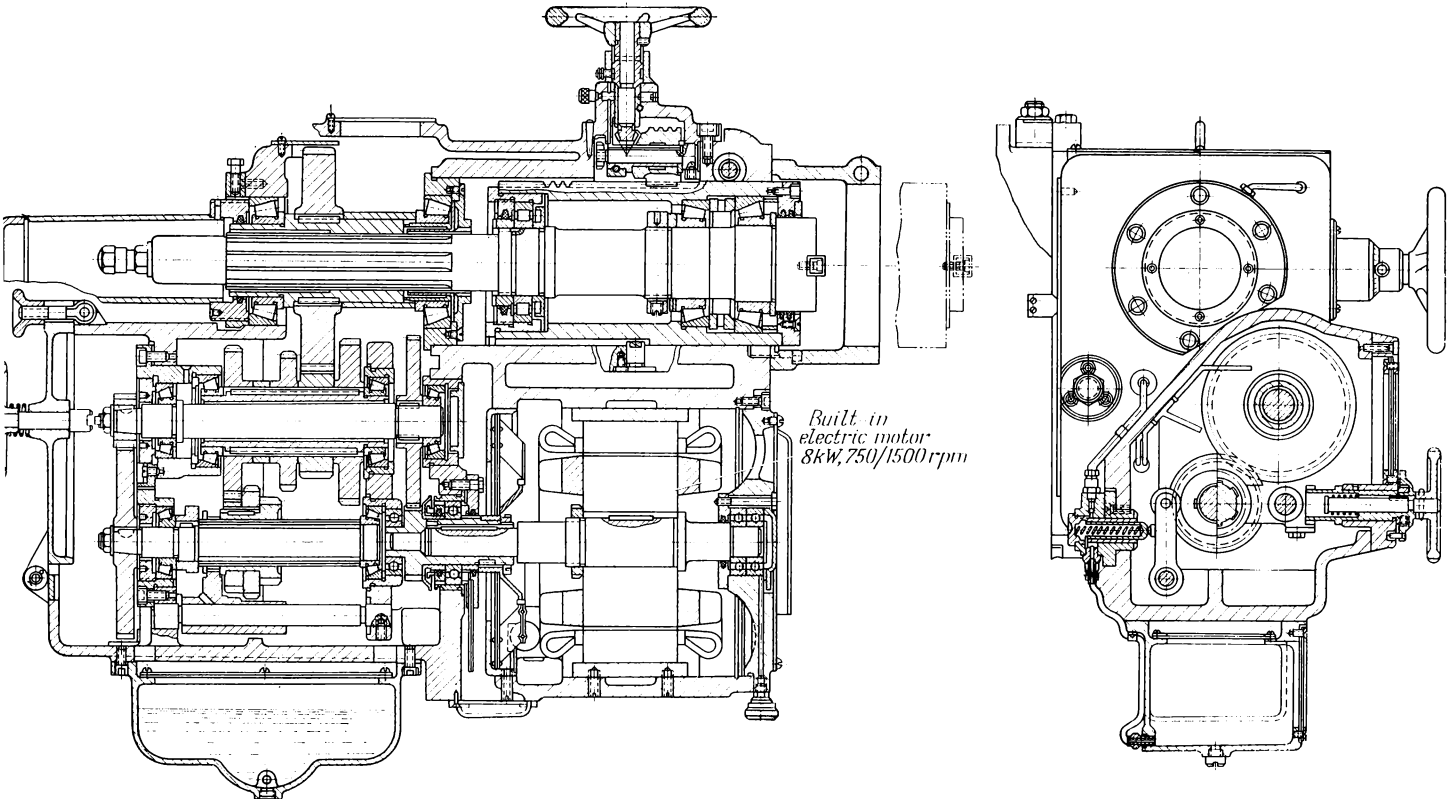


Fig. 33. Milling machine spindle head with a built-in two-speed electric motor, type MA 203KH 1/4 ВД

1. Spindle heads with reduced axial overall dimensions due, in some cases, to an increase in the radial overall dimensions (Fig. 33). Layouts of this type are suitable when the spindle head is to travel along ways perpendicular to the spindle axis. This design aims at reducing possible vibration resulting from the overhanging arrangement of the motor and other rotary masses of the drive (as in radial drills and planer-type milling machines). Such spindle heads are also used in vertical constructions of machine tools with a top drive for the spindle to reduce the overall height and to improve vibration-proof properties (see page 68 for features of the structure of such drives).

2. Spindle heads (or headstocks) with reduced radial overall dimensions due to an increase in the axial overall dimensions are used in heavy horizontal machine tools of the lathe group and others to reduce the transverse size of the machine tool and, consequently, the required width of the shop bays (see pp. 68 and 69).

3. Spindle heads (or headstocks) with a normal ratio between the axial and radial overall dimensions are used in most horizontal machine tools of the small and medium sizes.

The structure of their drive features the combined and moderate application of means for reducing axial and radial overall dimensions. Thus, one, and not two, linked gears are used to reduce axial overall dimensions. In the same manner, the axes of the shafts are made coincident in some but not all of the adjacent transmission groups.

II. Gearboxes with a divided drive. The spindle head or headstock and the gearbox may be designed as separate units connected by a belt transmission. The advantages of such a divided drive are: neither heat evolved by friction losses nor vibrations developed in the gearbox are transmitted to the spindle head (or headstock).

Further advantages, provided by adding a device to the above arrangement for relieving the spindle of the belt tension, are: (a) the spindle runs smoothly at high speeds, a feature of great importance in attaining a high class of surface finish and longer tool life in finishing operations; (b) with the provision of a multiplier device in the spindle head, two different types of drives to the spindle are obtained, one being a belt drive (with the spindle relieved of the belt tension) used for finishing operations at high spindle speeds n, and the other a geared drive for roughing purposes, thereby extending the speed range to make high-velocity machining feasible; and (c) in the preceding case, the multiplier device is disengaged to obtain the high steps of spindle speed, thereby increasing the efficiency of the drive and improving conditions for starting and braking the machine tool.

The pulley may be arranged at the end of the spindle, overhanging the bearing, to facilitate installation of the belt.

Methods for Changing Speeds in Gearboxes

The method used for engaging the various transmissions in a gearbox to change the speeds is determined mainly by the purpose of the machine tool and depends primarily upon the frequency with which the speeds are to be changed and the duration of the working movements.

If any of the speeds of the gearbox must be changed frequently, it is expedient for such changes to be made rapidly and while the machine is running. Thus, in small and medium-size turret lathes, the second extension group is changed over more often than other groups due to the alternation of drilling and reaming, or drilling or boring and thread cutting. Therefore, it is preferable to engage and disengage the transmissions of this group with friction clutches.

If the machining time for each operation element is small, it is desirable to change speeds quickly and without stopping the machine, so as to keep the relative share of handling time as small as possible.

I. Gearboxes with change (slip) gears. Speeds are changed by changing the gears of a group transmission between adjacent shafts with a constant centre-to-centre distance.

The advantages of the change-gear type of gearbox are:

1. It has small axial overall dimensions. The number of transmissions in a group is not restricted, insofar as design principles are concerned; it is limited only by the maximum and minimum permissible transmission ratios (see page 44) at the maximum range ratio of the group $R_{lim} = \frac{i_{max}}{i_{min}} = 8$ to 10.

2. If the required range ratio of the drive is within the indicated values, speeds can be changed by means of a single group transmission.

In this way, the structure and construction of a drive are simplified by using change gears (see, for example, the spindle drives of multiple-tool lathes).

3. With change gears it is impossible to make conflicting engagements so that no interlocking devices are needed.

In drives with inverse values of the transmission ratios $\left(i_j \text{ and } i_k = \frac{1}{i_j}\right)$, the same pair of change gears can be installed in the reverse order. This reduces the total amount of gears required for the group transmission. However, in drives with a great number of speed steps, when the series ratio φ is of small value, it is sometimes more expedient to reduce the number of change gears by the provision of two consecutive transmission groups with change gears instead of a single group.

The principal drawback of change gears is that a great deal of time is lost in changing speeds. To reduce this time loss, change gears may be installed on spline shafts or, less frequently, on tapered shaft journals with Woodruff keys (Fig. 33).

In the axial direction, change gears may be fixed on the shaft by quick-acting splined collars with locking devices or by C washers which can be readily removed to one side after slightly loosening the clamping nut or screw. The size of the nut or screw head is such that it freely passes through the bore of the change gear.

Change gears are often held in place by the shoulder of a cover fitted with small axial clearance. Another drawback of change gears on horizontal shafts is that the cover enclosing them is difficult to seal properly without a gasket. Housings with flanged walls are used, as well as oil-catch rings to prevent oil leakages in change gear arrangements.

Change gears are used when the spindle drive is to be changed over comparatively infrequently for different operations (and not operation elements) in mass and lot production, performed on automatic, semiautomatic, single-purpose and special production machine tools, and also for setting up general-purpose machine tools for a batch of workpieces.

II. Gearboxes with sliding gears. Group transmissions composed of sliding cluster gears can transmit high torque and power even if the drive is of comparatively small radial overall size. As in all geared transmission groups, the radial overall dimensions will be the minimum values for a given module when

$$i_{p\,max} i_{p\,min} = 1$$

where $i_{p\,max}$ and $i_{p\,min}$ are the maximum and minimum ratios of the transmissions in the group.

In gearboxes of this type, gears, not participating in the transmission of power to the spindle in a given engagement, are not in mesh and are consequently not subject to wear during this time.

These advantages have found speed changing with sliding cluster gears wide application in the gearboxes of machine tools, mainly general-purpose models, notwithstanding numerous shortcomings inherent in such arrangements. These include the following:

1. Speed changing is quite complicated and involves the disengagement of the gearbox drive, braking the gearbox shafts to obtain slow rotation, shifting the sliding cluster gears into and out of engagement, releasing the braking device and re-engaging the drive of the gearbox.

2. Breakdowns may occur when cluster gears are shifted into mesh when they are rotating too fast, or if two transmissions of a single group between adjacent shafts are simultaneously engaged. Provision must be made for

interlocking devices that prevent such conflicting engagements (see pages 260 and 262).

3. The group transmissions are of comparatively large axial overall size. The length of a group is $l \geqslant 2bp$, where p is the number of transmissions in the group and b is the face width of the gears in this group. A gear face width of $b = (4 \text{ to } 8)\ m$, where m is the module, is assigned to reduce the axial overall size.

4. The relatively large axial overall size does not permit more than $p = 4$ transmissions to be used in a group with sliding cluster gears. Groups with $p = 6$ transmissions are feasible only in rare cases when the components of the drive, adjacent to the group of such gears, are arranged outside of the gearbox housing.

This restriction in the number of transmissions in each group and the increased number of transmission groups complicate the structure of the spindle drive except in cases when the gear train required for reducing the motor speed to the minimum speed of the spindle must be made up of links whose number is sufficient to dispose the required number of transmission groups with sliding gears.

A long reducing gear train is commonly used in the spindle drives of general-purpose machine tools with a wide speed range. Sliding gears are the main type of group transmissions employed in the drives of such machines.

The large forces required to shift heavy cluster gears limit their application in manually controlled gearboxes of heavy machine tools.

Sliding gears are mounted on spline shafts; as a rule, spur gears are used.

III. Gearboxes with jaw clutches. The positive clutches used in the gearboxes of modern machine tools are often of the gear, or toothed, type which do not require fitting in manufacture and in which the forces of engagement are distributed more uniformly and over a greater number of working surfaces than in positive clutches of other types.

The advantages of changing speeds with jaw or other positive clutches are that they require small axial movement for engagement or disengagement, they permit helical or herringbone gears to be employed in the drive and they can be shifted with less effort than that needed to shift cluster gears. This last factor is of importance in the gearboxes of heavy machine tools.

Drawbacks associated with the use of jaw clutches for speed changing include: (a) the clutch teeth or jaws may be broken if engagement is made with the machine running and with a large difference in the rotational speeds of the clutch members, and (b) idle rotation of continuously meshing gears leads to friction losses between the gears and in their bearings on the shafts. To overcome the first of these drawbacks, it is necessary to disengage the

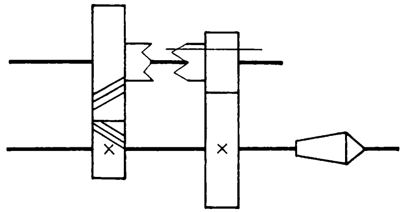


Fig. 34. Diagram of spindle train engagements

drive and to slow down the shafts carrying the clutch members, or to resort to a synchronizing device for reducing the difference in speed of the clutch members.

These friction losses substantially reduce the efficiency in certain gearboxes (with inverse speed steps) designed with sleeve bearings, and may cause self-braking in the gearing. Idle rotation of the gears, however, leads to losses even in gearboxes with antifriction bearings. These losses are especially prominent at higher steps of spindle speed.

The preceding shortcomings restrict the application of positive clutches in the gearboxes of up-to-date machine tools. Their most expedient use is in combination with sliding gears in an arrangement that excludes or limits idle rotation of the gears. Such devices include the back gearing arrangement in lathe headstocks in which the spindle is relieved of belt tension, and spindle drives designed according to Fig. 34 which exclude idle rotation of gears at the upper range of spindle speeds (Fig. 35).

Speed changing by means of positive clutches is sometimes used in the gearboxes of heavy machine tools to avoid the large forces required to shift heavy sliding cluster gears.

IV. Gearboxes with friction clutches. The possibility of rapid, smooth speed changing without stopping the machine makes the use of friction clutches for this purpose an effective method of reducing the time expended in handling a machine tool. Moreover, this arrangement permits helical and herringbone gears to be used in the gearbox.

The restriction in the torque that can be transmitted, the comparatively large axial and radial overall dimensions, the difficulties encountered in designing more than two transmissions in a group and more than three transmission groups in the gearbox, the high losses and wear in the idle rotation of continuously meshed gears, and the loss in efficiency due to friction in disengaged clutches are the main shortcomings of gearboxes in which speeds are changed by means of friction clutches. These drawbacks are sometimes supplemented by operating troubles, such as slipping and

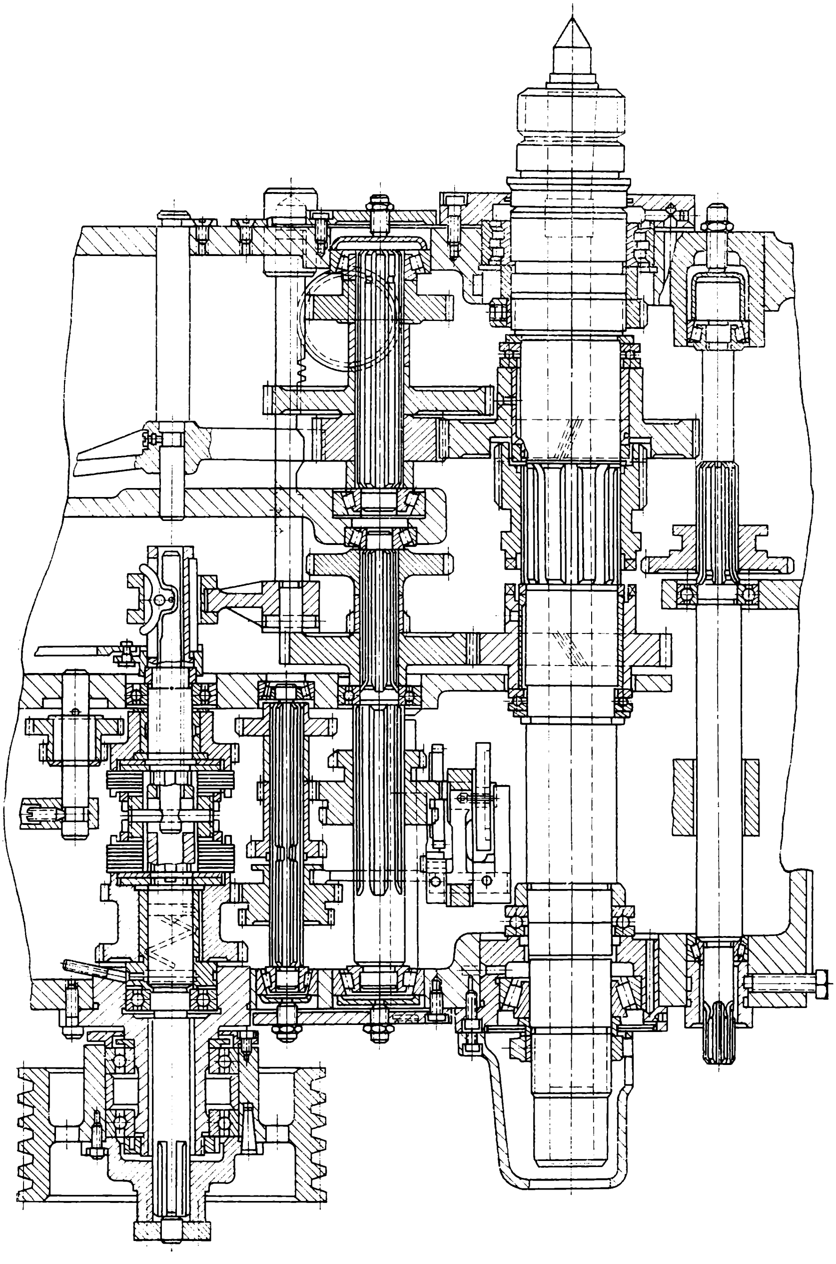


Fig. 35. Development of the speed gearbox of the model 1A62 lathe

overheating of the clutches, the necessity for frequent readjustments, and the transmission of heat from the clutches to the spindle unit with a consequent adverse effect on the machining accuracy.

Speed changing by means of friction clutches is used mainly in the group transmissions of small and medium-size turret lathes, and in some cases, in conjunction with a multiple-speed electric motor. In heavy turret lathes, engagements are made in the transmissions of the second extension group by means of friction clutches.

Of considerable promise are gearboxes equipped with electromagnetic disk clutches (see Fig. 32) or with magnetic-particle clutches in which the particles are in an oil slurry. These devices enable remote and automatic controls to be applied for gearboxes. An example is the model RT80P turret lathe, made in Hungary, in which the spindle speeds (n_1 = 28 or 35, n_{12} = = 1250 or 1600 and N = 14 kW) are changed by means of seven electromagnetic clutches controlled by a punched card or the dial of a preselector device.

Figure 36 illustrates the development of the speed gearbox of the model 1Б136 automatic turret lathe. This gearbox has six electromagnetic clutches (designed for 24V, d.c.). As can be seen in the drawing, three forward and three reverse speeds of spindle rotation are available for each pair of installed change gears A/B by making suitable engagements of the clutches.

The constructions of control devices for gearboxes are taken up in Chap. 11.

4-3. Feed Gearboxes

Principal Elements of Feed Mechanisms

The feed gearbox is a part of the feed mechanism which consists of the following separate elements:

1. The *drive of the feed mechanism* may be powered by a separate electric motor or from the spindle through a gear, chain or belt transmission. The most suitable type of transmission depends upon the maximum spindle speed n_{max}, the maximum torque $M_{t\ max}$ developed by the spindle in driving the feed mechanism and the required rigidity (torsional rigidity) of the kinematic chain between the spindle and the traversing element (feed rod).

If spindle speeds are infinitely variable, a feed drive originating at the spindle will maintain a constant feed per work revolution during the machining operation. For the same purpose, the electric motor that powers the feed drive is sometimes supplied from a generator whose rotor is linked directly to the shaft of the variable-speed electric motor of the spindle drive.

2. *Devices for engaging the feed mechanism*, in the form of jaw clutches, sliding gears or friction clutches, are arranged at the beginning of the feed

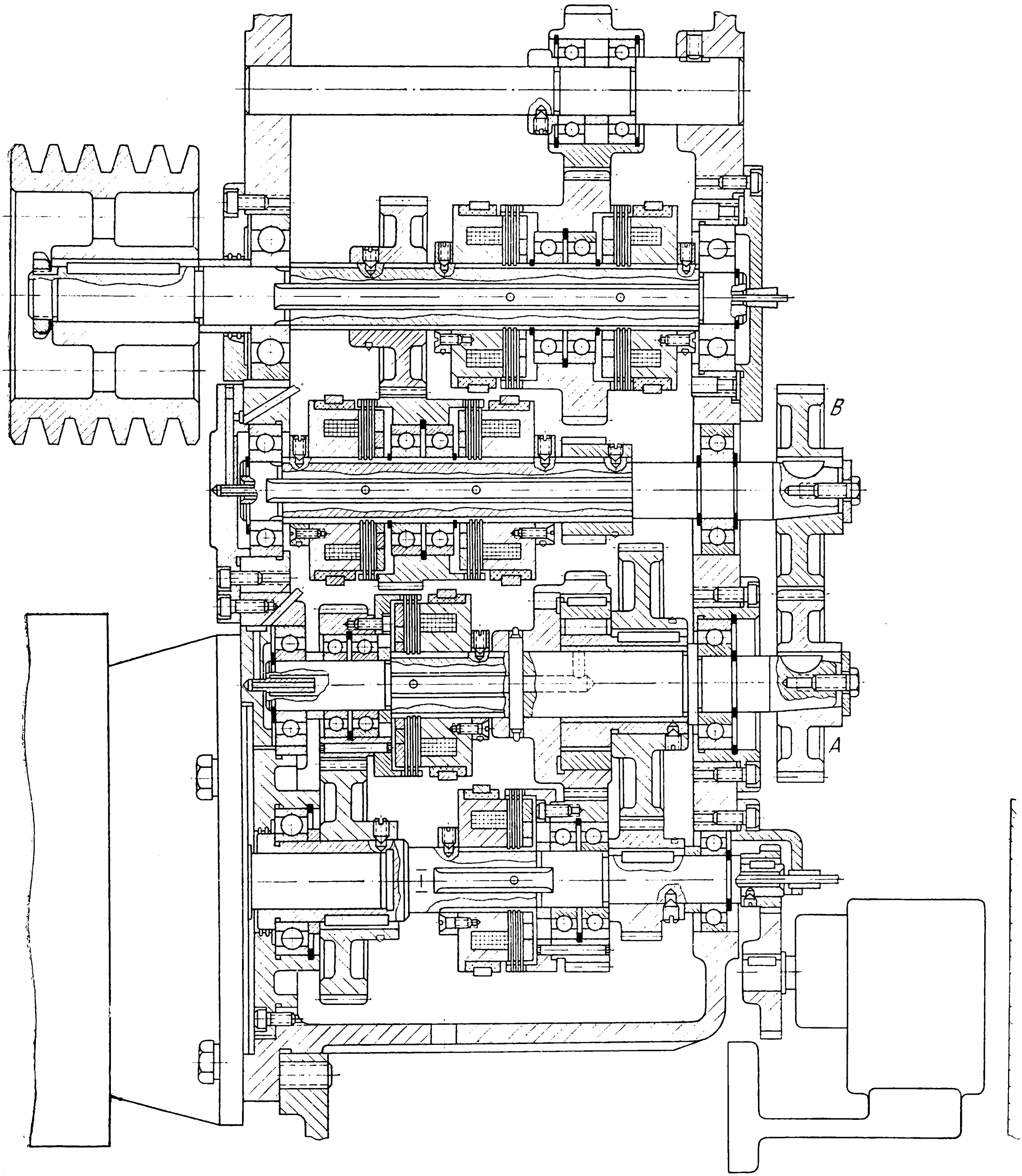
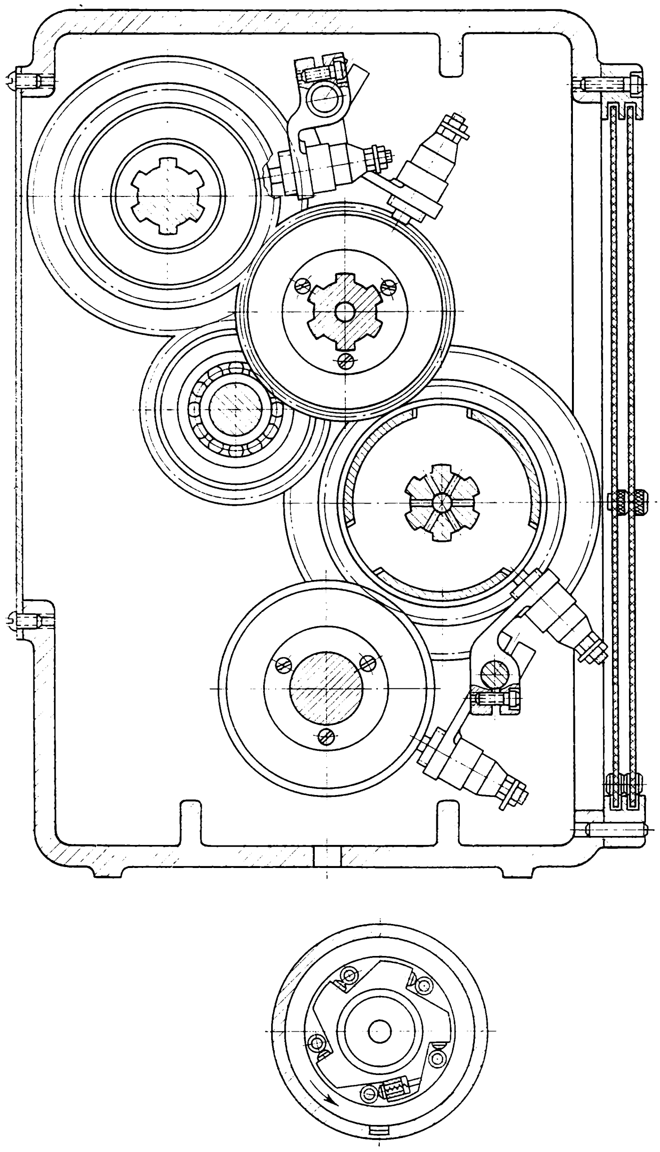


Fig. 36. Speed gearbox of the



..odel 1Б136 automatic turret lathe

train and where it begins to branch out in the working zone of the machine tool.

3. A *device for reversing the feed* is arranged in the working zone or is controlled from this zone.

4. A *safety device* for protecting the feed mechanism against overloads is provided in the part of the feed train where a variation in torque is due only to an increase in the feed force, i.e., between the traversing element (feed rod) and the last driven shaft of the feed gearbox.

5. *Single transmissions* of the working feed train, serving as speed reduction elements, are located between the feed gearbox and traversing element. Consequently, the feed gearbox shafts run at higher speeds and correspondingly lower torques. The single transmissions between the spindle and feed gearbox are required by the design layout of the feed mechanism as a whole.

6. A *gear train for rapid traverse movements* of carriages, tables, etc., powered by a separate electric motor or from the first shaft of the spindle drive, usually joins the train of working feeds at the end of the latter train, near the traversing element and following the single transmissions serving as speed reduction.

7. The *feed gearbox*, as mentioned above, is located at the beginning of the single transmissions for speed reduction with the aim of reducing the developed torques. On the other hand, it is desirable to arrange the feed gearbox nearer to the working zone, especially when the feeds are changed frequently, and when the feed rates of various carriages and slides are set up independently.

8. The *traversing element* of the feed mechanism. Its structural properties strongly affect the structure of the feed mechanism.

Certain Characteristics of the Feed Mechanism

Degree of speed reduction. If the feed drive is from the spindle, the total transmission ratio of the feed mechanism is determined from the equation

$$s = ti_{s/sp}$$

from which

$$i_{s/sp} = \frac{s}{t} \tag{99}$$

where s = rate of feed, mm per spindle revolution
t = pitch of the traversing element, mm
$i_{s/sp}$ = transmission ratio of the train from the spindle to the traversing element.

Here the pitch of the traversing element is taken to be the feed per revolution of the traversing element (lead of the thread on a lead screw, lead

of a cylinder cam, pitch of the Archimedian spiral on a plate cam, circumference of a rack pinion, etc.).

In cases when the feed drive is powered by a separate electric motor

$$s_m = n_{em} i_{s/e} t$$

Hence

$$i_{s/e} = \frac{s_m}{n_{em} t} \tag{100}$$

where s_m = rate of feed, mm per min
n_{em} = speed of feed drive motor, rpm
$i_{s/e}$ = transmission ratio of the train from the electric motor to the traversing element.

Therefore, at a given rate of feed, the pitch of the traversing element determines the degree of speed reduction and, consequently, the length of the feed gear train. The use of a lead screw with fine-pitch thread, for example, enables the feed mechanism to be designed with the shortest gear train, minimum transmission ratio and minimum speed reduction.

Work equation applied to the feed mechanism. The work equation can be conveniently employed to determine the torques developed by the various shafts of the feed mechanism. Thus

$$\frac{Qt}{2\pi} = \frac{M_{tj}}{i_{j/s}} \eta_{j/s} \tag{101}$$

where Q = feeding force
M_{tj} = torque of the j-th shaft
$i_{j/s} = \frac{n_s}{n_j}$ = transmission ratio from the j-th shaft to the shaft of the traversing element
$\eta_{j/s}$ = efficiency of the train from the j-th shaft to the traversing element
t as above.

Hence

$$M_{tj} = \frac{Qti_{j/s}}{2\pi\eta_{j/s}} = \frac{Q}{2\pi\eta_{j/s}} s_j \tag{102}$$

where $s_j = ti_{j/s}$ is the feed per revolution of the j-th shaft.

Thus, the torque of any shaft in the feed mechanism, other conditions being equal, is directly proportional to s_j, the feed per revolution of this shaft, or to $i_{j/s}$, the transmission ratio of the feed train from the given shaft to the shaft of the traversing element. It follows that, as distinguished from the spindle drive, the design train of the feed mechanism, i.e., the train

on which calculations should be based, is the train of maximum speed-increase transmissions. The maximum torque is developed by the slowest shaft of the train of maximum speed-increase transmissions, and not the slowest shaft in general.

Applying equation (102) to the spindle, we obtain

$$M_{t\ sp} = \frac{Qs}{2\pi\eta_{sp/s}} \tag{103}$$

where $M_{t\ sp}$ = torque developed by the spindle to drive the feed mechanism
$\eta_{sp/s}$ = efficiency of the train from the spindle to the shaft of the traversing element (feed rod, lead screw, etc.).

Equations (102) and (103) explain why in engine lathes, in which s_{max} may reach 200 mm per min and even more, the feed drive is designed more powerful than in multiple-tool lathes of the same size in which chain, or even belt, transmissions are used between the spindle and feed mechanism

When equation (102) is applied to the shaft of the traversing element, it becomes

$$M_{t\ s} = \frac{Qt}{2\pi\eta_s} \tag{104}$$

where $M_{t\ s}$ = torque developed by the shaft of the traversing element
η_s = efficiency of the traversing element.

Values of η_s for lead screws mating with ordinary nuts or with ball-bearing nuts are listed in Chap. 6, pages 146 through 149.

Notwithstanding the low efficiency of an ordinary screw and nut pair, subject to sliding friction, the torque developed by the lead screw, as well as by the shafts of the part of the feed train from the traversing element (i.e., lead screw) to the last driven shaft of the feed gearbox, is not very high in comparison with other types of traversing elements having large pitches. This, however, does not exclude the occurrence of very high torques in the part of the feed train from the spindle to the driving shaft of the feed gearbox, where the feeds per shaft revolution may reach very large values in cutting multiple-start worms and screws with large leads (multiple-start screws).

Requirements Made to Feed Gearboxes

Depending upon the purpose of the machine tool, various requirements may be made to the feed gearbox and to the feed mechanism as a whole in respect to (a) the number of feed steps, (b) range of feeds, (c) type of feed series (normally a geometrical series. but an approximately arithmetic

series for engine lathes), (d) nature of the feed motion (continuous or intermittent), (e) type of drive (from the spindle or from a separate electric motor), (f) absolute values of the rates of feed, (g) required accuracy of the feed rates, (h) permissible accumulated error of certain transmissions (in cutting high-precision threads, the kinematic chain must be as short as possible), (i) loads applied to the feed gearbox, and (j) frequency with which feeds are to be changed. These and certain other factors influence the structure and construction of feed gearboxes. This explains why such a great variety of designs are found in existing and new models of machine tools.

The rigidity of the kinematic chain between the spindle and the traversing element, the accuracy standards for the components of the gearbox and of the whole feed mechanism are assigned in accordance with the purpose of the machine tool, taking into account the effect of errors in manufacture and assembly of the mechanism on the machining accuracy of the machine tool being designed.

Manufacturing specifications applicable to components of speed gearboxes are suitable, in general, for the like components of feed gearboxes, including their housings.

4-4. Types of Feed Gearboxes

Feed gearboxes are classified in accordance with the type of geared mechanism they use to set up the feeds.

Feed Gearboxes with Change Gears on Fixed-Position Shafts

In conjunction with small axial overall size, such gearboxes provide a large number of feed rates, limited only by the maximum and minimum permissible transmission ratios. They find application in machine tools requiring infrequent changes of feed, such as automatic, semiautomatic, single-purpose and special machine tools employed in lot and mass produc, tion.

Feed Gearboxes with Sliding Gears

Gearboxes of this type are more suitable for frequent feed changing than gearboxes with change gears, and are therefore widely used in general-purpose machine tools (Fig. 37). Their capability of transmitting high torques (in comparison with other designs given below) and of operating at high speeds

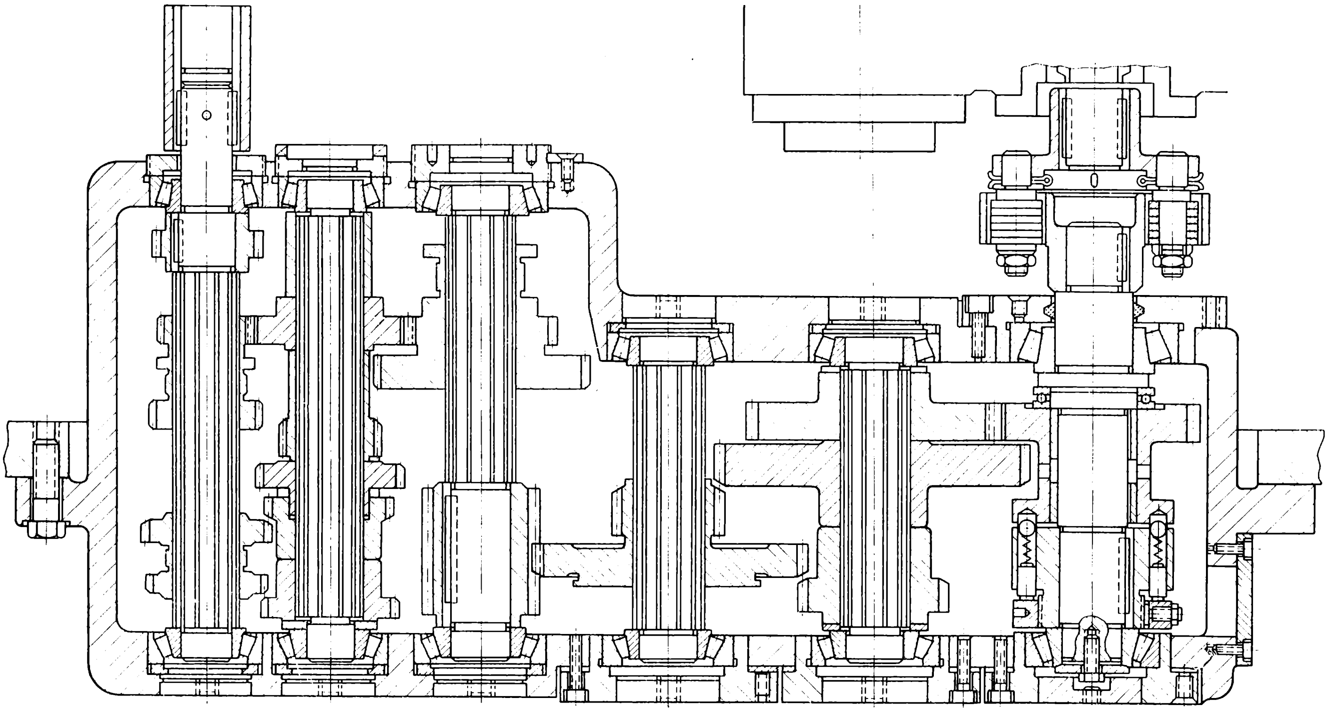


Fig. 37. Feed gearbox of a planer-type milling machine

without idly rotating mated gears enable these gearboxes to be applied efficiently in feed drives powered by a separate electric motor, as well as in heavy lathes, milling machines and vertical boring mills, and various high-speed machine tools.

A drawback is the practical impossibility of using helical gears to obtain a series of exact transmission ratios.

Feed Gearboxes with Intermeshing Gear Cones and Sliding Keys

The compact arrangement of this design, the feasibility of arranging up to 8 or 10 transmissions in a single group, the possibility of using helical gears to obtain a series of exact transmission ratios, and the control of all the engagements of a pair of cones with a single lever are the main advantages of these gearboxes (Fig. 38).

Possible cocking of the sliding key, insufficient rigidity of the key shaft which is weakened by the longitudinal slot and excessive speeds of gear rotation if the key shaft is the driving member are drawbacks excluding the use of these gearboxes for transmitting high torques and at high speeds of the shafts.

The sliding-key type of feed gearbox finds application in small and, in some cases, medium-size drill presses and turret lathes.

The insufficiently reliable location of the narrow gears on the shafts limits the diameter of the gears that can be used. For this reason, sliding-key mechanisms are usually employed as the main transmission group of feed gearboxes.

Feed Gearboxes with Gear Cone and Tumbler Gear (Norton Type)

The design of this type of gearbox, due to the provision of the idle tumbler gear (z_0 in Fig. 39), does not need to comply with the general condition that the sums of teeth of all the pairs of mating gears on adjacent shafts are equal. This extends the possibilities for obtaining more precise transmission ratios, this being a feature of great value in setting up the feeds of engine lathes.

Using a single gear (z in Fig. 39) on one of the shafts to obtain all the transmissions, the range ratio of the transmission group $R_{lim} \leqslant 4$. This is a small value for a feed step series based on a geometrical structure. An arithmetic series, one convenient for cutting standard threads, provides as many as 10 or 11 feed steps for the same range without resorting to an

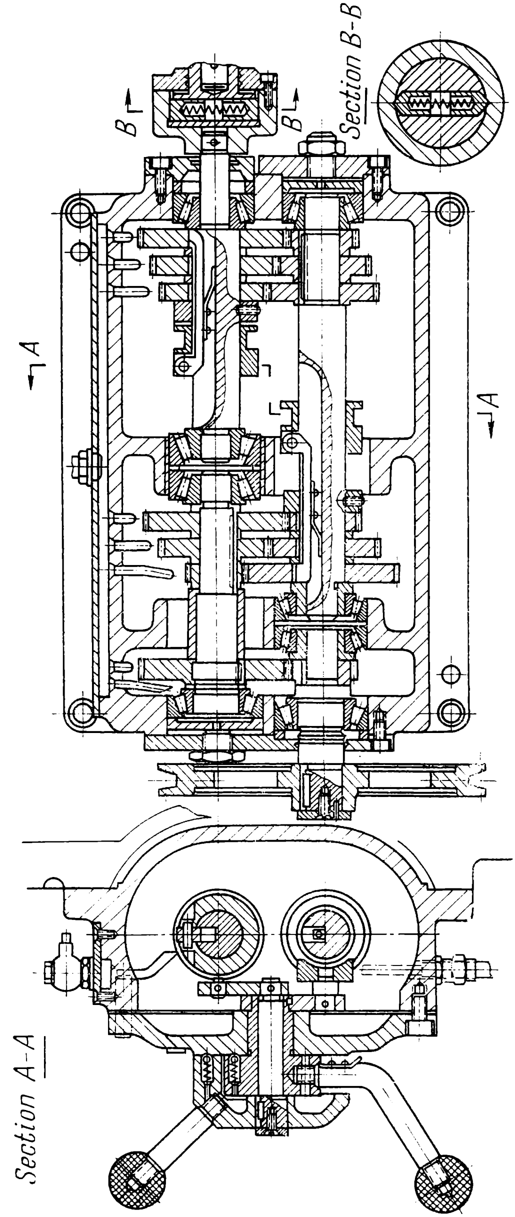


Fig. 38. Feed gearbox of the model 1330 turret lathe

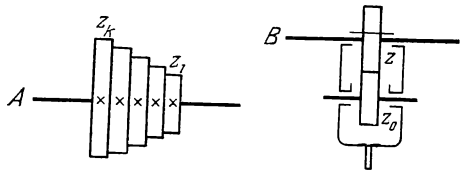


Fig. 39. Kinematic diagram of a gear cone and tumbler gear gearbox

axial overall size that exceeds all acceptable limits. This relatively large number of feeds are changed by a single lever which both shifts and locks the tumbler-gear arm.

The common application of Norton feed gearboxes in engine lathes is due to these features.

One important advantage of these gearboxes is that they require a small number of gears ($K + 2$ gears for K transmissions). Serious disadvantages are the insufficiently rigid and accurate meshing of engaged gears, unreliable lubrication, and the possibility of dirt getting into the gearbox through the slots in the housing. These drawbacks are overcome in up-to-date machine tools by using closed gearboxes and by making efforts to increase the rigidity of the mounting of the tumbler-gear arm, as has been done in the closed-type feed gearbox of the model 1A62 lathe (Fig. 40).

It is expedient to include a Norton-type gearbox in the train of the feed mechanism in such a way that motion is transmitted from the gear cone to the tumbler-gear arm shaft in cutting metric threads, and in the opposite direction in cutting inch threads. Then the number of teeth of the gears in the cone will be directly proportional to the metric thread pitches and threads per inch (inch threads). Indeed, in transmission from the gear cone to the tumbler-gear arm shaft (see Fig. 39)

$$s_j = C_1 \frac{z_j}{z} t$$

from which

$$z_j = \frac{z}{C_1 t} s_j \tag{105}$$

In transmission from the tumbler-gear arm shaft to the gear cone

$$s_j = \frac{25.4}{n_j} = C_2 \frac{z}{z_j} t$$

whence

$$z_j = \frac{C_2 z t}{25.4} n_j \tag{106}$$

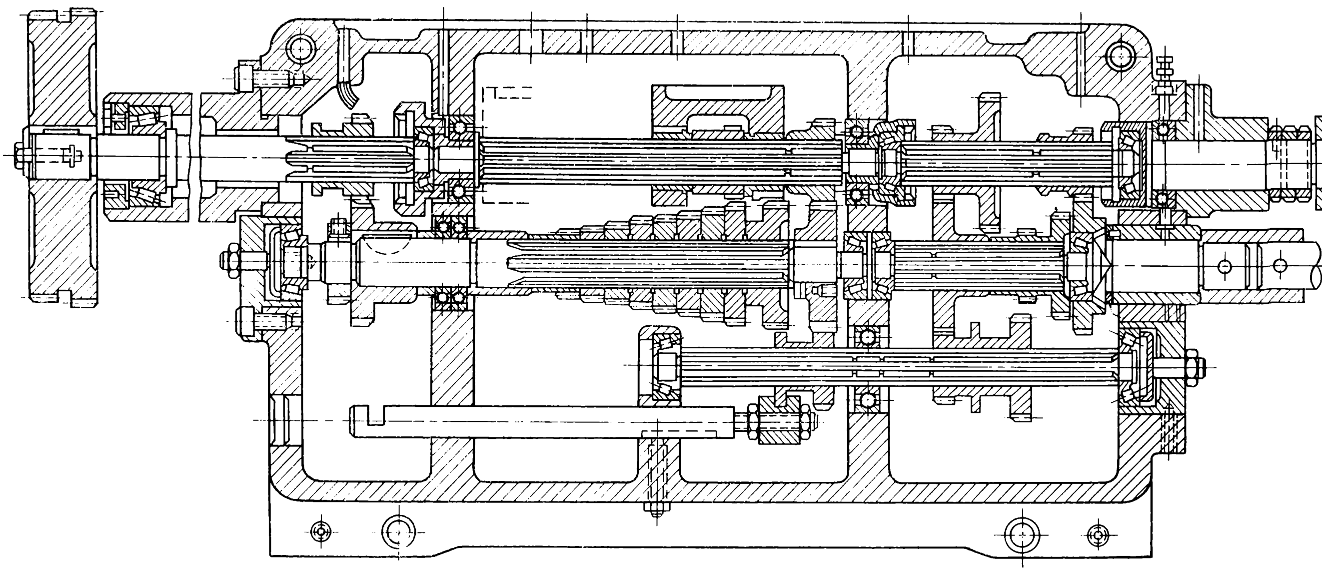


Fig. 40. Feed gearbox of the model 1A62 engine lathe

where C_1 and C_2 = constant transmission ratios of the corresponding kinematic trains

s_j = feed in mm per spindle revolution, equal to the thread pitch, mm

n_j = threads per inch of the inch thread

t = pitch of the lead screw, mm

z = number of teeth on the sliding gear

z_j = number of teeth of the cone gear engaged by the tumbler gear.

Feed Gearboxes Designed as Change-Gear Quadrants

A set of change gears enables feeds to be set up with any degree of accuracy. Given sufficient distance between the shafts linked by the set of change gears, transmission ratios up to $i_{min} = \frac{1}{8}$ can be applied. This increases the

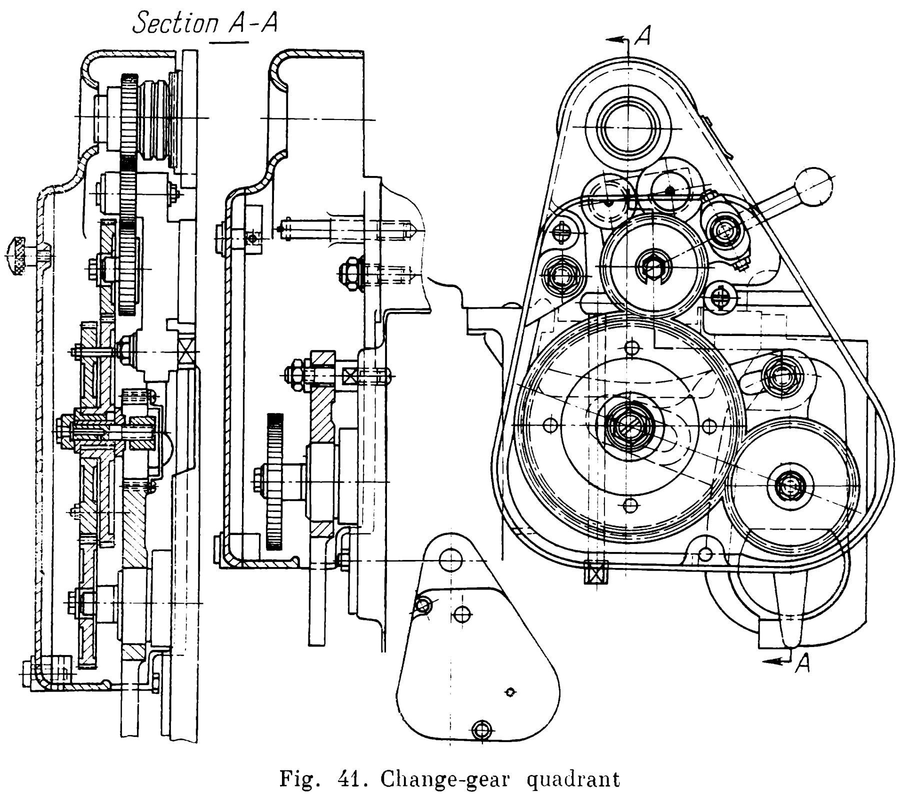


Fig. 41. Change-gear quadrant

feed range ratio and simplifies the structure and construction of the feed drive. The adjustable quadrant makes it possible to compensate for errors in the positions of the axes of the shafts being linked together.

These properties of change-gear quadrants render them suitable for various types of machine tools, especially those intended for lot and mass production.

Such change-gear arrangements are widely applied in thread- and gear-cutting machines (see Part Two). They also enable an engine lathe to operate with a short train, bypassing the feed gearbox, as is required to increase the accuracy of thread being cut.

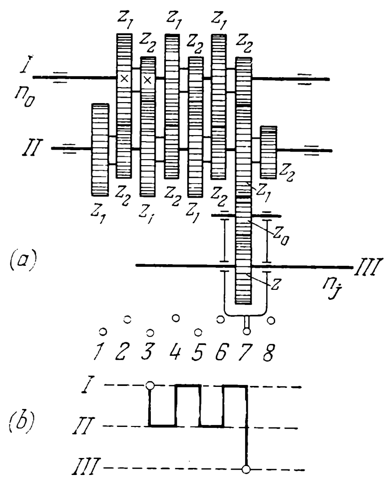


Fig. 42. Diagram of a Meander drive with a tumbler gear

Figure 41 illustrates the change-gear unit of an engine lathe.

One or two pairs of change gears is sufficient, in the great majority of cases, to obtain the required rates of feed. Three pairs of change gears are resorted to only in rare cases when especially low transmission ratios are required, or these ratios must be set up with exceptional accuracy (to obtain, for example, thread pitches). An example is the model 1810 relieving lathe which has three pairs of change gears in the feed train.

Feed Gearboxes of the Meander Type

A Meander drive is a three-shaft mechanism made up of a series of identical double-cluster gears and a sliding carrier with a tumbler gear (Fig. 42*a*). Such features as the single-lever controls, small axial overall size and wide range ratio enable this drive to be conveniently used as the first extension group of the feed mechanism. It is extensively employed in engine lathes for this purpose.

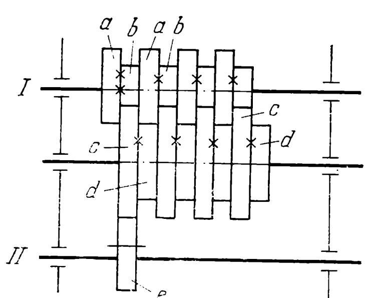


Fig. 43. Diagram of a Meander drive with a sliding gear

In addition to drawbacks due to the use of a tumbler gear (see page 95), in a Meander drive all the cluster gears rotate continuously in mesh, including cluster gears which do not participate in a particular engagement.

In some designs, the tumbler gear is replaced by a sliding gear (Fig. 43) which engages only the larger gears of the clusters.

This construction possesses increased rigidity but more cluster gears are required to obtain the same number of feed steps.

Attempts to mount the cluster gears on ball or roller bearings have led to complications in the construction. A three-shaft gearbox with sliding gears is preferred, for this reason, by designers for high-speed machine tools, even though this may lead to a more complex system of controls.

In the Meander drive, shown schematically in Fig. 42*a*, rotation is transmitted through various numbers of return steps (Fig. 42*b*), in accordance with the position of the carrier and tumbler gear. Introducing the notation $c = \frac{z_1}{z'}$, we obtain the following series of transmission ratios (see Fig. 42)

$$i_1 = \frac{z_1}{z_2} \times \frac{z_1}{z'} = c\frac{z_1}{z_2} = c\left(\frac{z_2}{z_1}\right)^{-1}; \quad i_2 = \frac{z_1}{z_2} \times \frac{z_2}{z'} = \frac{z_1}{z'} = c;$$

$$i_3 = \frac{z_2}{z_1} \times \frac{z_1}{z'} = c\left(\frac{z_2}{z_1}\right)^{1}; \quad i_4 = \frac{z_2}{z_1} \times \frac{z_2}{z'} = c\left(\frac{z_2}{z_1}\right)^{2}$$

$$i_5 = \frac{z_2}{z_1} \times \frac{z_2}{z_1} \times \frac{z_2}{z_1} \times \frac{z_1}{z'} = c\left(\frac{z_2}{z_1}\right)^{3}, \text{ etc.}$$

As a rule, in engine lathes $z = z'$, i.e., $c = 1$ and $z_1 = 2z_2$. The series of transmission ratios in this case is $\frac{2}{1}, \frac{1}{1}, \frac{1}{2}, \frac{1}{4}, \ldots$.

4-5. Rapid Traverse Mechanisms

In modern machine tools, especially those operating with an automatic cycle, idle travel movements of the working members—tables, carriages, heads, etc.—are carried out at a higher speed to increase the production capacity of the machine. The speed of rapid traverse movements obtained by a mechanical drive ranges from 2 to 12 m per min and, most often, from 4 to 8 m per min.

The structure of the rapid traverse mechanism is determined by the properties of its following main elements: traversing element of the feed drive, drive of the rapid traverse train, and the device for joining the rapid traverse and working feed trains.

It is known from the theory of mechanisms and machinery that the use of cam mechanisms as the traversing element enables the rate of feed to be varied within a single cycle and reversed. This property of cam mechanisms permits us to do without a rapid traverse train in small automatic screw machines.

A rapid traverse train and a device for its reversal are required to obtain rapid forward and reverse traverse in machine tools having a traversing element with constant pitch (screw and nut, pinion and rack, etc.). This

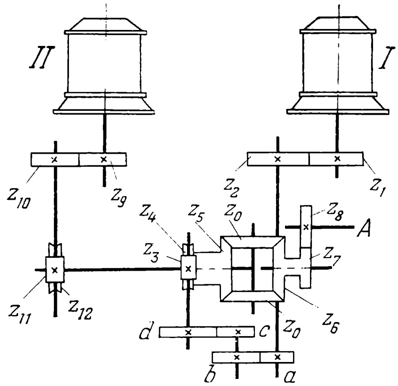


Fig. 44. Kinematic diagram of a rapid traverse drive

train is powered either from a high-speed shaft, running at constant speed, at the beginning of the drive train of the machine, or from a separate electric motor, a reversible motor if necessary. The use of an individual electric motor simplifies the structure and control of rapid traverse movements.

The working feed and rapid traverse trains are joined in most machine tools by means of a single- or two-direction (reversible) overrunning clutch. With this arrangement, the working feed train is not disengaged when the rapid traverse drive is engaged, and the former is automatically engaged when the latter is disengaged. This substantially simplifies the control system (see, for example, the kinematic schemes of automatic and semi-automatic machine tools in Part Six).

Similar structural properties are possessed by differentials which are better suited than overrunning clutches to carry the high inertia loads that occur in reversing large masses in rapid traverse motions, when traversing elements with a constant pitch are employed in heavy machine tools.

When electric motor I of the working feed drive is switched on, in the arrangement shown schematically in Fig. 44, the transmission ratio from its shaft to shaft A will be

$$i_{wf} = \frac{z_1}{z_2} \times \frac{a}{b} \times \frac{c}{d} \times \frac{z_3}{z_4} \times 1 \times \frac{z_7}{z_8} \tag{107}$$

since the shaft of the planet gears is held stationary by the self-locking worm gearing.

When electric motor II of the rapid traverse drive is switched on, the transmission ratio of the train will be

$$i_{rt} = \frac{z_9}{z_{10}} \times \frac{z_{11}}{z_{12}} \times 2 \times \frac{z_7}{z_8} \tag{108}$$

In this case, bevel gear z_5 of the differential is fixed, the planet gears roll around it and the transmission ratio of the differential is $i_{dif} = 2$.

A twin jaw clutch is more seldom used in up-to-date models to join the working feed and rapid traverse trains since, in this case, the construction of the feed drive and the control system become more complicated.

In many cases braking devices are required in the feed drive to avoid overtravel after disengaging rapid traverse if the traversing elements are not of the self-braking type.

Rapid traverse movements can be accomplished much more simply and efficiently by means of a hydraulic drive (see Vol. 2, Part Four).

4-6. Infinitely Variable Drives in Machine Tools

Advantages of Infinitely Variable Drives in Operation

Infinitely variable (also called variable-speed) main and feed drives have found considerable application in modern metal-cutting machine tools. Their main advantages are the possibility of setting up the optimum cutting conditions (speed and feed) with higher accuracy than with a stepped drive and, what is more essential in practice, the possibility of changing speeds of the main drives or feeds without stopping the machine. As a result, the operator can set up or find the most expedient machining conditions for each job. In turning stepped shafts and irregular contours, facing ends and in cutting off, an infinitely variable speed drive enables a constant cutting speed to be maintained (if the variable-speed drive is automatically controlled) by varying the angular velocity in accordance with cross travel of the slide.

A constant cutting speed, in these cases, leads to an increase, not only in the output, but in the tool life as well, especially for cutting tools tipped with ceramics or cemented carbides (which are susceptible to changes in cutting speed). A more uniform quality of surface finish is attained. The possibility of operating at optimum cutting speed is essential because at higher speeds the additional output does not reimburse the extra time lost in changing tools and the cost of extra tool replacements, while at lower-than-optimum speeds the output drops (as does the tool life for certain cemented carbides).

Easy and smooth speed changing without stopping the machine enables operation beyond the limits of the zone of resonance vibrations.

Variable-speed friction drives, used in machine tools, operate much more quietly than gear or chain drives; in normal condition and properly maintained, such drives are practically noiseless.

In machine tools requiring a fine graduation of speed steps ($\varphi = 1.06$ or 1.12), the construction of the drive is frequently more compact and less expensive if a simple gearbox is used in conjunction with a variable-speed device, instead of a complicated multiple-step gearbox of the same type.

Beginning with the progression ratio $\varphi = 1.26$, the substitution of stepless for stepped speed variation offers a material gain in cutting speed (or rate of feed) and a consequent reduction in machining time. The possibility of rapidly changing speeds without stopping the machine ensures a savings of handling time in machine operation. Thus, the use of a variable-speed drive promotes an increase in the production capacity of a machine tool.

Types of Infinitely Variable Drives

Various methods are in use for stepless variation of the rates of the working motions. The selection of the method to use in any particular case depends upon the purpose of the machine (general-purpose, specialized or special; for roughing, finishing or microfinishing); the power required for cutting and the required type of mechanical characteristic; the required range ratio; the permissible increase in the cost of the machine; etc. Each of the possible solutions—electrical, hydraulic, mechanical or combined speed variation—has its specific operational advantages and disadvantages and, consequently, its field of preferable application.

Stepless Electrical Speed and Feed Drives

Electrical variation is accomplished by varying the speed of the electric motor which drives the corresponding train of the machine tool.

D. C. electric motors with shunt adjustment are used chiefly in heavy machine tools. Most convenient, in this case, is a generator-motor drive with a range ratio of $R = 10$ to 15.

The use of rotary amplifiers in a generator-motor set (Ward-Leonard system) enables the speed range to be substantially extended. These drives are suitable for machine tools requiring very large range ratios, in the order of 500, 1000 and higher.

Stepless electrical speed and feed drives can be readily automated.

The main drawbacks of these systems are the comparatively large overall size and costs.

Stepless Hydraulic Speed and Feed Drives

Hydraulic drives are widely used to obtain infinitely variable rates of rectilinear motion in machine tools. In most cases, this refers to feed drives,

but in some machine tools (planers, shapers, slotters and broaching machines) main drive speeds are varied in this way.

The essential advantages of a hydraulic drive in obtaining stepless speeds are: the wide range of speed variation, possibility of rapidly changing both the magnitude and direction of the speeds, smooth reversal, convenience of remote control and its automation, automatic protection against overloads, and self-lubrication of the system.

A drawback of hydraulic drives is the insufficiently flat characteristic curve resulting from leakages and the effect of the temperature on the oil viscosity. At low speeds ($v = 12$ to 15 mm per min) the operation of a hydraulic drive becomes unstable.

A hydraulic drive is rarely used for rotary motion in a machine tool because of its high cost and low efficiency after wear. Here, it gives way to other types of infinitely variable drives.

Speed variation in the hydraulic systems of machine tools is considered in detail in Part Four, Vol. 2.

Infinitely Variable Mechanical (Friction) Drives

Most mechanical variable-speed drives employed in machine tools are of the friction type and therefore their operation involves friction losses.

The following types of friction losses may be distinguished:

(a) Losses due to unfavourable kinematic conditions in the contact zone which lead to a difference in the velocities of the conjugate points of the working surfaces. The kinematic losses due to friction are reduced when the working surfaces in the contact zone approximate the shape of two cylinders with parallel axes or two cones with a common apex.

(b) Losses due to distortion of the working surfaces in the contact zone. These losses are not very high (2 or 3 per cent) and are reduced when the Young's modulus of the material of the contacting members is increased, for example, by replacing plastics with steel.

(c) Losses due to slipping of the working members of the variable-speed drive, similar to slip in belt drives. Slip losses are increased when the reserve adhesive force is reduced between the working members due to variations in the cutting forces or to the influence of inertia forces in starting and reversing the drive, etc.

As mentioned above, the possibility of changing the speed of the output shaft without stopping the machine is a highly advantageous feature of mechanical variable-speed devices, but it is usually associated with the impossibility or difficulty of setting the speed when the variable-speed device is not running.

Other drawbacks of mechanical stepless speed adjustment are the non-rigid kinematic characteristics of friction-type variable-speed drives and the variations in the maximum transmitted power when the speed is changed in most variable-speed drives.

Methods of Extending the Range of Infinitely Variable Speeds in Drives

The range ratio of mechanical variable-speed drives depends upon the principle and construction of the device, and may range from $R_{vs} \cong 2$ to 4 (variable-speed drives with wide V-belts and adjustable sheaves) to $R_{vs} \cong 10$ to 25 (chain- and ball-type variable-speed drives). In most cases $R_{vs} = 4$ to 6. Such a range ratio enables the main and first extension groups of stepped transmissions (see page 65) to be replaced by variable-speed devices. Therefore, the use of devices designed to obtain stepless speeds considerably simplifies the structure of the drive (in the same way as groups with change gears). This is another advantage of variable-speed drives.

Although in some cases $R_{vs} \gg 4$ to 6, the range ratio R for the whole drive may be such that the variable-speed device must be supplemented by a stepped multiplier device with a range ratio R_{st} that complies with the condition

$$R_{vs}R_{st} = R \tag{109}$$

On the other hand, equation (62), i.e.,

$$\varphi_p = R_{gb}\varphi$$

can be written, for a combination of a variable-speed device and a stepped speed gearbox when $\varphi \longrightarrow 1$ for the series of spindle speeds, as

$$\varphi_p = R_{gb} \tag{110}$$

In a stepped multiplier type of gearbox, consisting of transmission groups from the first to the last extension group, the progression ratio of the series of transmission ratios is:

for the first group

$$\varphi_2 = R_{vs}$$

for the second group

$$\varphi_3 = R_{vs}R_2 = R_{vs}\varphi_2^{p_2-1} = R_{vs}^{p_2} = \varphi_2^{p_2}$$

and for the third group

$$\varphi_4 = R_{vs}R_2R_3 = R_{vs}\varphi_2^{p_2-1}\varphi_3^{p_3-1} = R_{vs}R_{vs}^{p_2-1}R_{vs}^{p_2(p_3-1)} = R_{vs}^{p_2p_3} = \varphi_2^{p_2p_3}$$

It is evident that the stepped multiplier type of gearbox is to be set

up in the same way as an ordinary gearbox with a progression ratio

$$\varphi' = R_{vs} \tag{111}$$

in which case

$$R_{st} = \frac{R}{R_{vs}} = \varphi'^{(z-1)} = R_{vs}^{z-1} \tag{112}$$

Due to the variable slip in electric motors, belt drives and variable-speed friction drives, the actual range ratio of a variable-speed device may turn out to be less than R_{vs}. Therefore, to avoid gaps in the stepless speed series, the progression ratio is taken equal to $\varphi' = (0.94 \text{ to } 0.96)\ R_{vs}$.

Sometimes, to obtain a convenient whole number of speed steps z in the multiplier type of gearbox, it may be necessary to reduce the ratio to $\varphi' < R_{vs}$ and to permit overlapping of the stepless series in changing over the speeds of the gearbox. On the other hand, the transmission ratio is sometimes taken as $\varphi' > R_{vs}$, allowing gaps in the stepless series of spindle speeds, to simplify the structure of the drive and to reduce the number of steps z.

Let us assume, for example, that $R = 60$, $R_{vs} = 3$ and $R' = \frac{R}{R_{vs}} = \frac{60}{3} = 20$. It is impossible to assign $z = 3$ because the value $R' = 20$ is far in excess of the permissible limiting range ratio of a single group transmission. If we take $z = 4 = 2 \times 2$, then $R' = \varphi'^{z-1} = \varphi'^{3}$, from which

$$\varphi' = \sqrt[3]{R'} = \sqrt[3]{20} \cong 2.71 < R_{vs} = 3$$

Here, the stepless series overlaps in changing over the transmission of the gearbox.

Let us assume that $R = 48$ and $R_{vs} = 6$. Then $R' = \frac{R}{R_{vs}} = \frac{48}{6} = 8$. This range ratio can be obtained by one extension group. To simplify the structure of the drive, we take the number of transmissions of this group to be $p' = z = 2$ and obtain $R' = 8 = \varphi'^{z-1} = \varphi' > R_{vs} = 6$. This ratio leads to a gap in the stepless series. In order that the ratio $\varphi' < R_{vs}$ it is necessary to take a number of transmissions $p' = z = 3$, but this complicates the structure of the drive.

Constructions of Mechanical Variable-Speed Drives for Machine Tools

A great many kinds of mechanical variable-speed drives, mostly of the friction type, are employed in machine tools. These devices may differ both in construction and in their operational parameters which include: the type, power rating N_{em} and speed n_{em} of the drive electric motor; maximum and minimum speeds, n_{max} and n_{min}, of the output (driven) shaft, and consequently, the range ratio R; values of the required power N_{max} and N_{min} on the input shaft at the speeds n_{max} and n_{min}; efficiency values η, overall

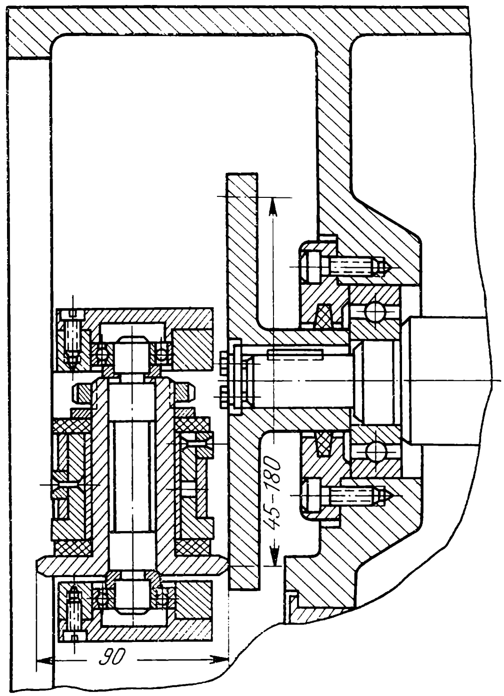


Fig. 45. Face roller variable-speed drive of the model КУ-39 heavy shaper

dimensions and the weight. By comparing the specifications, the most suitable type and size of variable-speed drive can be selected for the machine tool being designed.

Of the large number of models of mechanical variable-speed drives, the following find application in up-to-date machine tools:

1. Variable-speed devices based on direct contact between the driving and driven elements, including face roller drives with either roller or disk as the driving element, and variable-speed drives in which the speeds are changed by adjusting the angle between the axes of disks due to the swivel of a hinge-mounted electric motor. Illustrated in Fig. 45, as an example, is the face roller variable-speed drive in the feed train of a heavy shaper, model КУ-39, made by the Kolomensk Heavy Machine Tool Plant. The feed drive of the ram toolslide consists of an induction electric motor, three-step feed gearbox with a range ratio of $R_{gb} = 16$, and a variable-speed friction drive with a range ratio of $R_{vs} = 4$. Thus, the required range ratio of feed variation $R_s = 64$ is obtained.

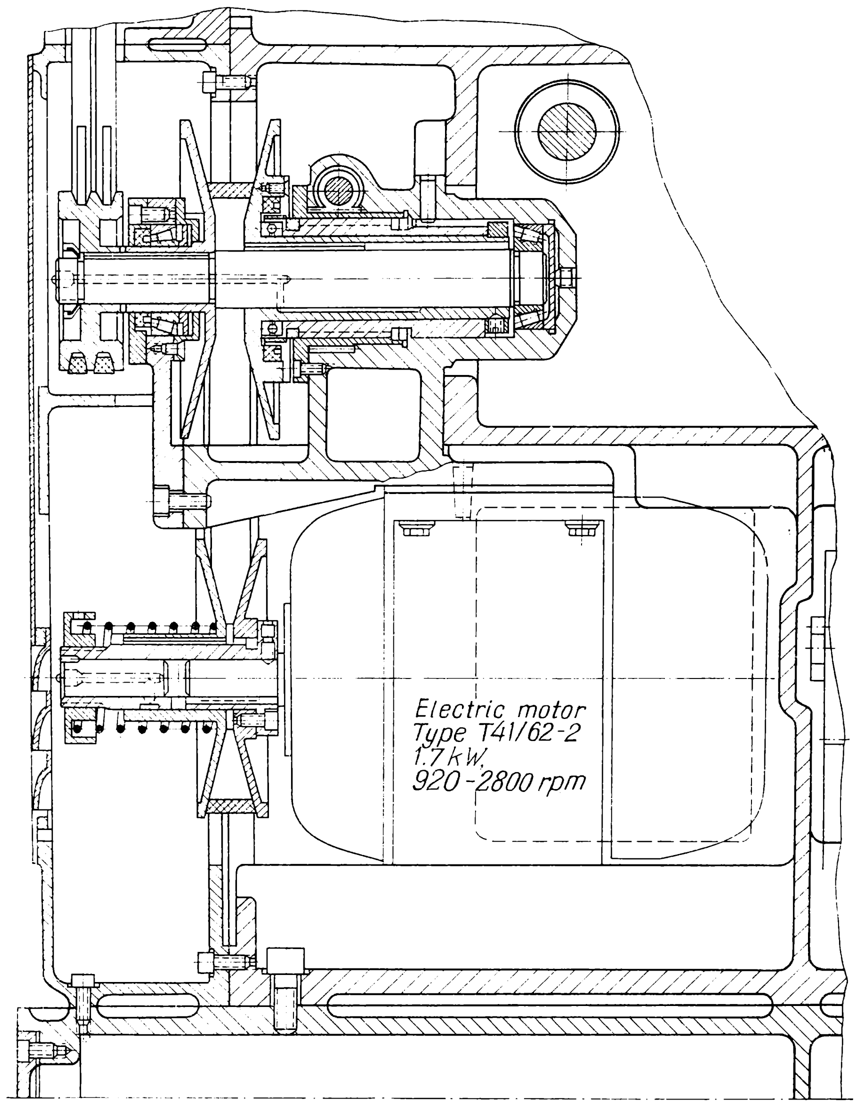


Fig. 46. Longitudinal section of a variable-speed device built into the main drive of the model 5K301 gear hobber

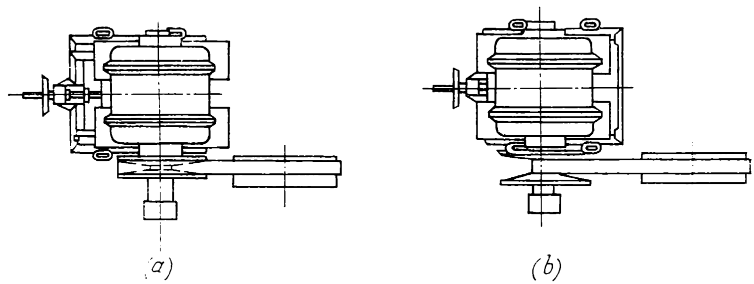


Fig. 47. Variable-speed drive with one adjustable-pitch sheave

2. Variable-speed drives with two pairs of adjustable conical sheaves. The sheaves are so made that the two halves can be moved axially to change the pitch (effective) diameter. A wide V-belt runs over the two adjustable-pitch sheaves. A size range of these variable-speed drives was developed by ENIMS. They have output shaft speed range ratios $R_n = 4$ ($n_{max}/n_{min} = 3000/750$ or $2000/500$) and $R_n = 2$ ($n_{max}/n_{min} = 3000/1500$). The power required on the input shaft varies from 2.8 to 7 kW for the various sizes. Some versions operate with constant power or with constant torque on the output shaft. Shown in Fig. 46 is the longitudinal section of such a device built into the main drive of the model 5K301 gear hobber.

In some designs of this group, only one of the sheaves is adjustable, while the other is of the ordinary type (Fig. 47*a* and *b*).

3. Variable-speed drives with two adjustable-pitch sheaves linked together by a steel ring instead of a belt. This type of device is built into the headstock of a cylindrical grinder as shown in Fig. 48. It has also been employed in the feed drive of the model 2B440 jig borer.

4. Toroidal variable-speed drive, developed by V. Svetozarov of TSNIITMASH (Central Research Institute of the Heavy Engineering Industries). This principle was used by the Krasny Proletary Plant in the design of one model of engine lathe (Fig. 49).

A unique variable-pitch chain drive has found some application in machine tool engineering. Known as the P.I.V. (positive, infinitely variable) drive, it consists of two variable-pitch sheaves with grooved conical faces and a self-tooth-forming chain meshing with the grooves of the sheaves. The sheaves are adjusted axially to change the pitch diameters as in the V-belt type of drive. Such drives have not been used in Soviet machine tools.

The construction of variable-speed drives of various types, and design procedures are described in textbooks on Machine Design while design formulas can be found in reference books.

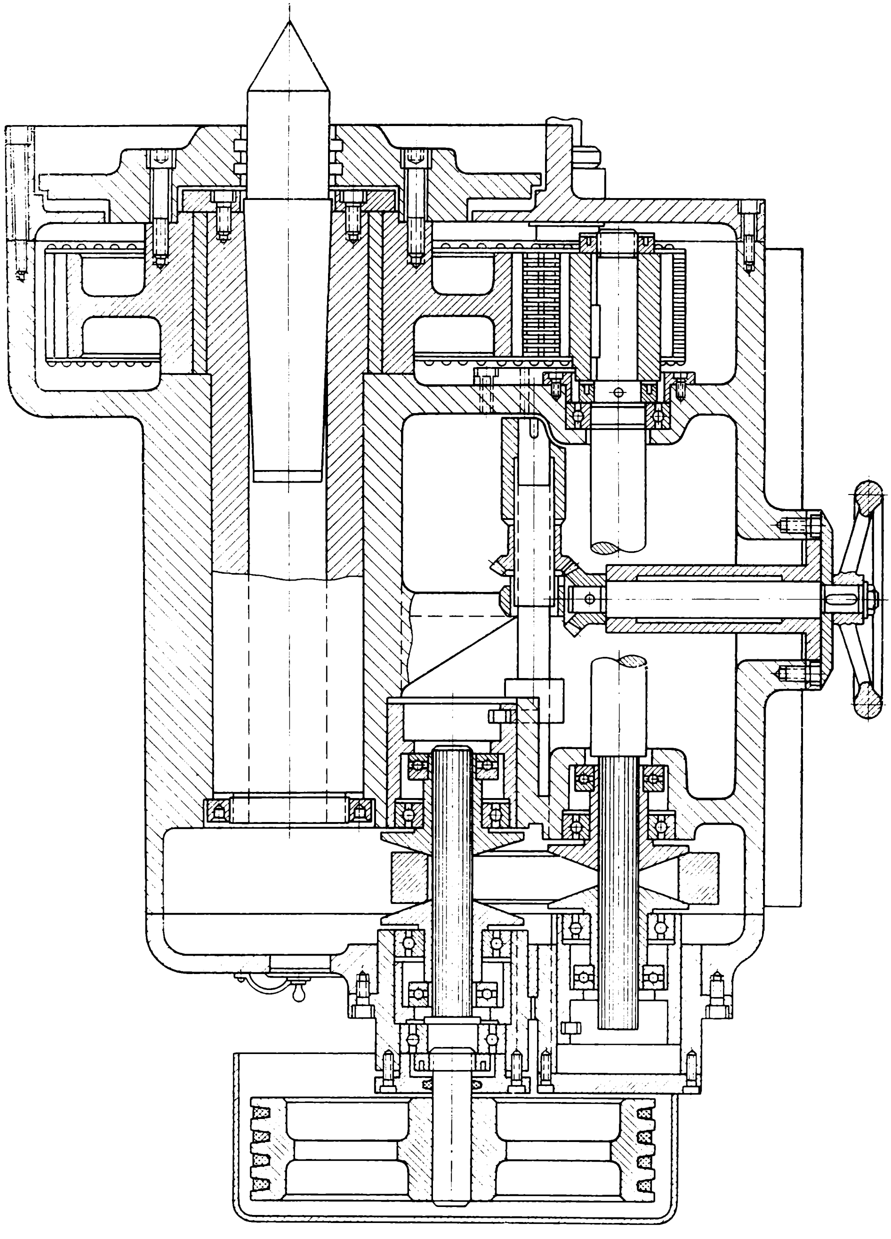


Fig. 48. Headstock of a cylindrical grinder

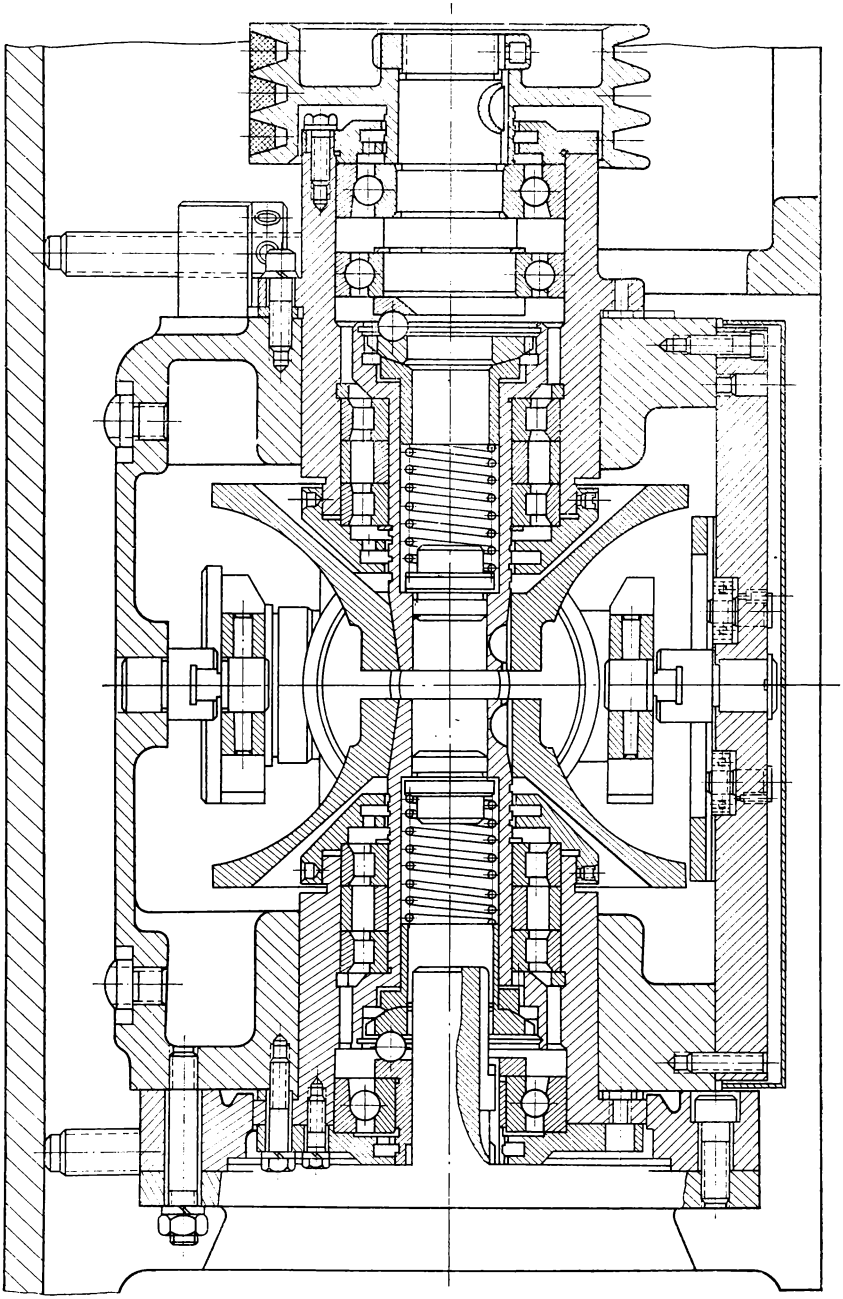


Fig. 49. Toroidal variable-speed drive, developed by V. Svetozarov of TSNIITMASH and incorporated in the design of an engine lathe manufactured by the Krasny Proletary Plant

CHAPTER 5

SPINDLES AND SPINDLE BEARINGS

5-1. Principal Requirements Made to Spindle Units

Machining accuracy depends to a considerable degree in many types of machine tools upon the rotational accuracy of the spindle which transmits motion to the cutting tool or to the work. This imposes the following principal requirements on the spindle units of machine tools:

1. *Rotational accuracy* is usually characterized by the runout of the front end (nose) of the spindle. The permissible spindle nose runout values, both radial and axial, have been standardized for most types of general-purpose machine tools. In designing special machine tools, these values are assigned on the basis of the required accuracy of the workpieces that are to be machined.
2. *Rigidity* is determined as the capacity of the spindle to retain its correct position when acted on by various working forces. Excessive deformation of the spindle has a detrimental effect on the machining accuracy and on the service life of the spindle bearings and drive.
3. *Vibration-proof properties* should be possessed by the spindles of high-speed machine tools, especially those intended for performing finishing operations.
4. *Wear resistance* of the bearing surfaces is required in cases when the spindle runs in sleeve bearings or when there is relative longitudinal motion of elements of the drive and the spindle (as in drilling, boring and other machines).

These requirements are complied with by correctly selecting the materials and construction of the spindle and its bearings.

5-2. Materials and Construction of Spindles

The main requirement made to the great majority of spindles is sufficient rigidity which depends, in part, upon the Young's modulus of the spindle material. Since the Young's modulus of various steels is practically the same, there are no grounds for using alloy steels for spindles, unless their application is dictated by other requirements. Therefore, the spindles of Soviet machine tools are usually made of medium-carbon structural steel 45

which subsequently undergoes a heat treatment known as structural improvement (quenching followed by high tempering to a hardness of 22-28R_C).

If above-standard requirements are made to the spindle, and its surfaces (or parts of the surfaces) must have a high hardness, steel 40X is sometimes used with a heat treatment consisting of quenching followed by tempering to a hardness of 40-50R_C. Better results can be obtained by employing induction hardening which can provide a surface hardness of 48-60R_C on the spindle journals (for sleeve bearings) and subject the spindle to much less distortion in heat treatment. Low-carbon casehardening steel, type 20X, is used in cases when very high surface hardness of the spindle journals is required. Here, the heat treatment consists of carburization, quenching and tempering to a hardness of 56-62R_C.

Spindles of high-precision machine tools, not subject to heavy loads, are made of steel 35XMЮA which undergoes nitriding followed by quenching and tempering to a hardness of DPH 850-1000. Nitriding provides an exceptionally high surface hardness with very little deformation.

Spindles of heavy machine tools are made of manganese steel, type 50Г2, with subsequent normalization (spindles subject to low loads) or hardening followed by high tempering to a hardness of 28-35R_C.

In specific cases, hollow spindles of large diameter for horizontal boring and other machines can be expediently made of grey cast iron, grade СЧ15-32 or СЧ21-40, or of high-strength nodular cast iron.

TABLE 2

Principal Types of Machine Tool Spindle Noses

<table>
<tr><th>Design</th><th>Applications</th><th>USSR Std</th></tr>
<tr><td>1</td><td>Lathes</td><td>OST 428</td></tr>
<tr><td>2</td><td>Turret and multiple-tool lathes, grinders, etc.</td><td>GOST 2570-58</td></tr>
<tr><td>3</td><td>Milling machines</td><td>GOST 836-62</td></tr>
<tr><td>4</td><td>Drilling and boring machines</td><td>GOST 2701-44</td></tr>
<tr><td>5</td><td rowspan="2">Grinders</td><td>GOST 2323-51</td></tr>
<tr><td>6</td><td>GOST 2324-43</td></tr>
</table>

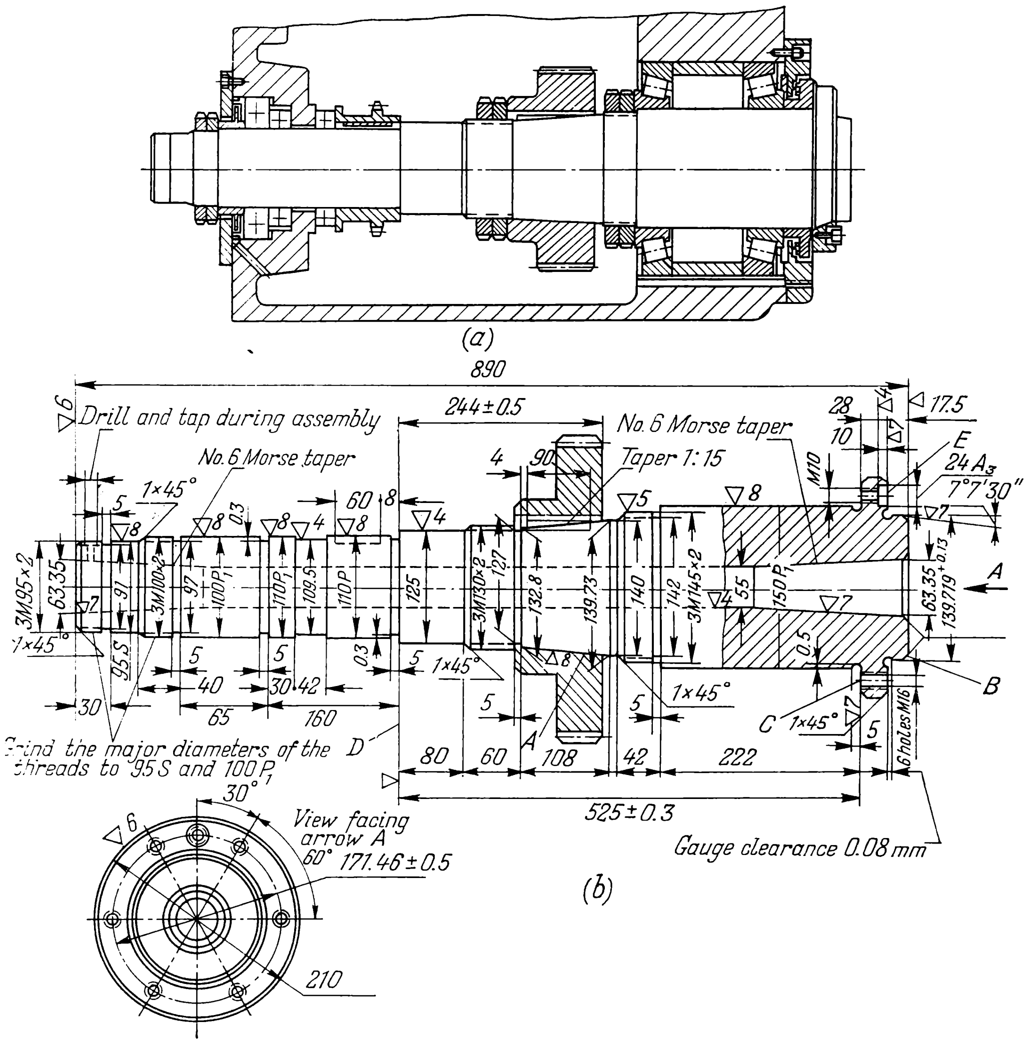


Fig. 50. Machine tool spindle:
(a) spindle unit; (b) working drawing

The design features of a spindle depend upon the kind of cutting tool or workpiece it is to carry, the fits of the elements of the drive and the type of bearings it is to run in (see, for example, Fig. 50).

Spindle noses of general-purpose machine tools have been standardized (Table 2). This, to a considerable extent, predetermines the construction of the spindle as a whole.

5-3. Spindle Design

Rigidity calculations involve the determination of the deflection in bending and, in some cases, the twist in torsion.

In working out the design diagram, the spindle is usually replaced by a beam on hinged supports. Such an assumption is valid when there is one ball or roller bearing in each support. In more exact calculations, several ball or roller bearings in a single support are to be regarded as an elastic support, while a spindle running in sleeve bearings is regarded as a beam on an elastic foundation. This last case can also be conditionally reduced to a beam on hinged supports to which a reactive moment M is applied in the support (Fig. 51). The value of this moment varies, according to experimental data, from zero (for insignificant loads, as in finishing machines) to 0.3-0.35 of the external moment acting in the middle section of the spindle on the support.

In tentative calculations, bending deflection and torsional twist of spindle sections can be analytically determined. For example, in the design diagram in Fig. 51, the deflection at the spindle nose and the slope in the front support are

$$y = \frac{1}{3EI}\left[P_1a^2(a+l) - 0.5P_2abl\left(1-\frac{b^2}{l^2}\right) - Mal\right] \tag{113}$$

and

$$\theta = \frac{1}{3EI}\left[P_1al - 0.5P_2al\left(1-\frac{b^2}{l^2}\right) - Ml\right] \tag{114}$$

where I = average value of the moment of inertia of the sections of the spindle

$M \leqslant 0.35P_1a$.

It proves expedient, in refined calculations, to construct the complete elastic line of the spindle axis, using a semigraphical method for this purpose.

Taking into account the elastic strain of the supports, the deflection at the nose of a two-support spindle, relieved of the action of the drive (Fig. 52), can be determined by the formula

$$y = \frac{P}{j} = P\left[\frac{1}{kj_0} + \frac{1}{j_{0.h}} + \frac{(1+k)^2}{j_B} + \frac{k^2}{j_A}\right] \tag{115}$$

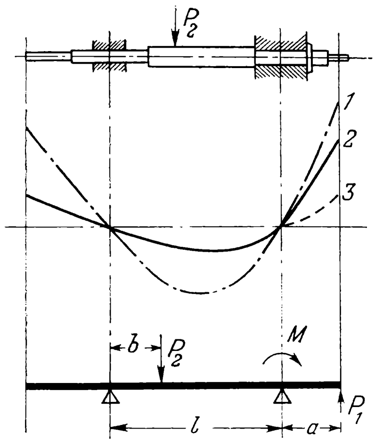


Fig. 51. Spindle deflection diagram:
1—theoretical elastic line of the spindle axis with the spindle on knife-edge supports; 2—actual elastic line of the spindle axis; 3—theoretical elastic line of the spindle axis with the spindle fixed in the front support

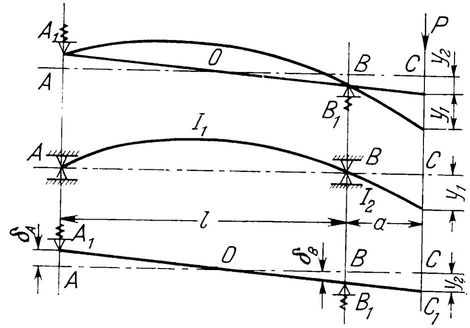


Fig. 52. Diagram for spindle rigidity calculations with account for the unit deflection of the supports

where j = rigidity of the spindle unit

$j_0 = \frac{3EI_1}{a^3}$ = conditional rigidity of the spindle in the length between the supports

$j_{0\,oh} = \frac{3EI_2}{a^3}$ = conditional rigidity of the overhanging part of the spindle

$k = \frac{a}{l}$ = ratio of the overhanging part to the length between supports

P = load applied at the spindle nose

I_1 and I_2 = average moments of inertia of the sections of the spindle between the supports and in the overhanging part, respectively

j_B and j_A = rigidity of the front and rear spindle supports, respectively

E = Young's modulus of the spindle material.

If, instead of rigidity j in the last equation, we substitute its reciprocal, unit deflection $c = \frac{1}{j}$, we obtain

$$y = P\left[\frac{1}{k}c_0 + c_{0oh} + (1+k)^2 c_B + k^2 c_A\right] \tag{115a}$$

which is more convenient for calculations.

The detailed form of this equation is

$$y = P\left[\frac{a^2}{3E}\left(\frac{l}{I_1}+\frac{a}{I_2}\right)+(1+k)^2 c_B + k^2 c_A\right] \quad \textbf{(115b)}$$

Using this equation, the optimum distance l between the spindle supports, as regards the rigidity of the spindle unit, can be readily determined.

The permissible deflection at the spindle nose should take into consideration the requirements made to the machining accuracy of the machine tool. As a rule, this deflection is limited to a certain fraction $\left(\text{usually } \frac{1}{3}\right)$ of the tolerance for runout at the spindle nose. Deflection and slope in other sections along the length of the spindle are limited by requirements of proper operation of the transmission and bearings. According to the investigations of D. Reshetov, for example, toothed gearing operates normally if the angle by which the gear axes are out-of-parallel does not exceed

$$\theta_{max} \leqslant \frac{CP}{10^4 b^2} \text{ rad} \quad \textbf{(116)}$$

where P = peripheral force, kgf
b = face width of the gears, mm
C = coefficient taking into account the nature of load distribution along the gear teeth ($C \simeq 5$ to 15).

The following values of permissible deflection and slope angle are used as tentative norms in machine tool engineering practice:

$$y_{max} \leqslant 0.0002l \quad \theta_{max} \leqslant 0.001 \text{ rad} \quad \textbf{(117)}$$

where l is the distance between the supports (see Fig. 52).

In the design of a spindle on which the rotor of an electric motor is mounted, the maximum deflection between the supports is limited by the value

$$y_{max} \leqslant 0.1\delta \quad \textbf{(118)}$$

where δ is the average width of the air-gap clearance between the rotor and the stator of the built-in motor.

In this last case, in addition to other forces, the load due to unilateral magnetic attraction is also taken into consideration.

Vibration behaviour calculations, including the determination of the natural frequency of the spindle to avoid resonance vibrations, are advisably to be carried out for high-speed spindles.

The natural frequency of vibrations can be determined by any of the methods given in the course of Theoretical Mechanics. In cases when no considerable masses overhang the supports, it proves expedient to use the graphical method (Fig. 53—for a turret lathe spindle).

These calculations are carried out in the following sequence. The elastic line of the spindle axis is constructed representing the deflection due to

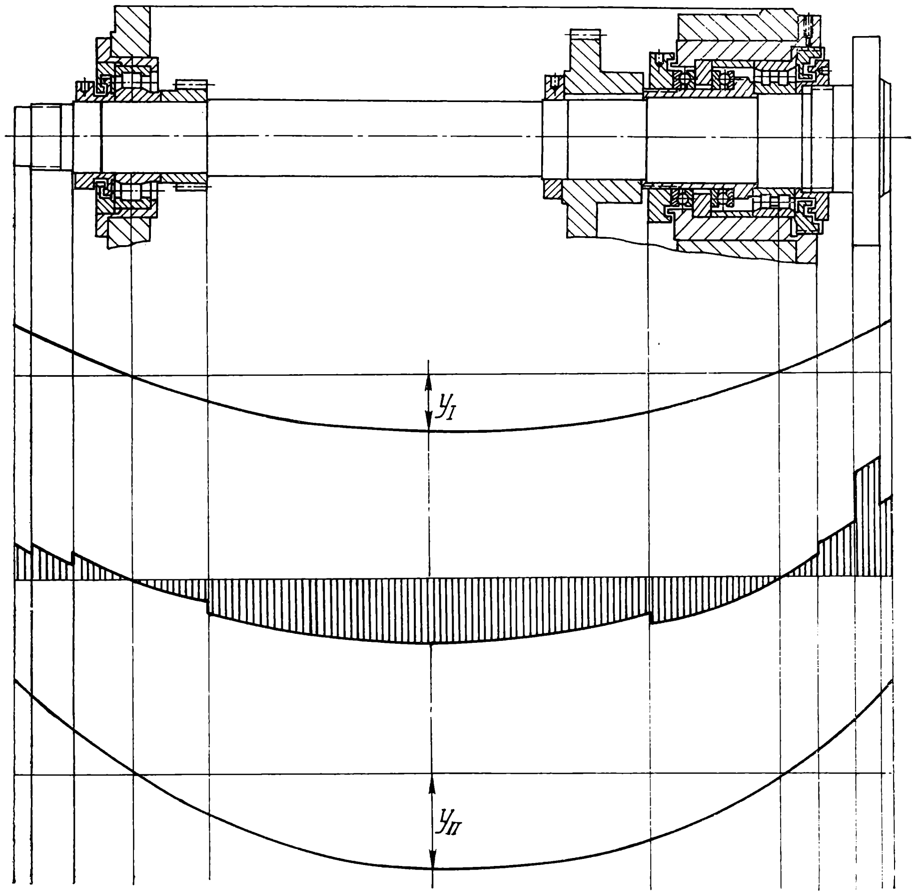


Fig. 53. Diagram for vibration behaviour calculations of a spindle

the dead weight of the spindle. Then a random angular velocity ω_0 of spindle rotation is assumed, and a new elastic line is constructed representing the deflection under the action of centrifugal forces at each section of the spindle along its length. Thus

$$A_x = F_x \rho \omega_0^2 y_x \tag{119}$$

where F_x = cross-sectional area of the spindle
ρ = mass density of the spindle material
y_x = deflection at the given cross section.

This method is based on the fact that elastic lines, constructed with a sufficient degree of accuracy, are geometrically similar, i.e., $y_I = \text{const}\ y_{II}$ (see Fig. 53), and the critical angular velocity is determined from the relationship

$$\omega_{cr} = \omega_0 \sqrt{\frac{y_{II}}{y_I}} \qquad (120)$$

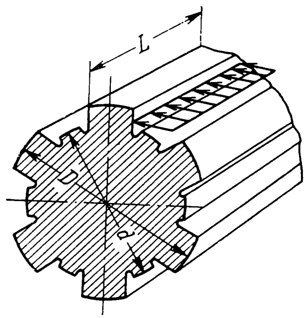


Fig. 54. Diagram for specific pressure calculations for splined sections

The following condition is usually laid down to eliminate the danger of resonance

$$\frac{|\omega_{cr} - \omega|}{\omega} \geqslant 0.25 \text{ to } 0.3$$

where ω is the maximum angular velocity of spindle rotation.

Strength calculations are used in checking heavily loaded spindles. This involves checking the factor of safety n for alternating stresses by the formula

$$n = \frac{(1 - \xi^4)\, d^3 \sigma_{-1}}{10 \sqrt{(aM)^2 + (a_k M_t)^2}} \qquad (121)$$

where d = outside diameter of the spindle

$\xi = \frac{d_0}{d}$ = ratio of the inside to the outside diameter of the spindle

σ_{-1} = endurance limit for bending with a symmetrical cycle of stresses

M and M_t = average values of the bending moment and torque.

Coefficients a and a_k take into consideration stress concentration and the degree of variation of the moments (and torques). They are determined by the relationships

$$a = k_\sigma (1 + C) \quad \text{and} \quad a_k = \frac{\sigma_{-1}}{\sigma_T} + k_\tau C_k \qquad (122)$$

where k_σ and k_τ = dynamic coefficients of stress concentration for normal and shearing stresses (for tentative calculations $k_\sigma \cong k_\tau \cong 1.7$ to 2)

σ_T = yield stress.

The coefficients $C = \frac{M_a}{M}$ and $C_k = \frac{M_{ta}}{M_t}$ are determined as the ratio of the amplitude of the moment or torque to its average value and hence depend upon the type of machining performed on the machine tool. For example, for microfinishing operations $C \cong C_k \cong 0$; for finish turning and drilling $C \cong C_k \cong 0.1$ to 0.2; and for milling and roughing operations in which an extremely nonuniform allowance is removed $C \cong C_k \cong 0.3$ to 0.5. The safety factor is usually taken as $n = 1.3$ to 1.5.

Besides the types of calculation given above, the specific pressure should

hecked on the surfaces of splined sections of spindles (Fig. 54). Here

$$p=\frac{8M_t}{(D^2-d^2)\,Lz\psi} \qquad (123)$$

re M_t = maximum value of the torque
nd d = major and minor diameters of the spline shaft
L = length of the fitting
z = number of splines
ψ = factor taking into account nonuniform utilization of the spline surfaces, due to errors in manufacture; usually $\psi = 0.75$.

ermissible design values of the specific pressures are listed below; if spindle is not heat treated, these values should be halved.

Type of spline fitting	Permissible design specific pressure, kgf/mm²
Fixed	12 to 20
Movable, but not under load	4 to 7
Movable under load	1 to 2

5-4. Spindle Bearings

he following specific requirements are made to the spindle bearings of hine tools:

Accuracy of guidance (radial and axial) of the spindle; accordingly, only ll clearances are permitted in spindle bearings in conjunction with rigidity of the bearings.

Adaptability to variable operating conditions; in many machine s the spindle bearings are subject to various loads in a wide range of ds. and with frequent starting and stopping.

her requirements, common to all shaft bearings, including those of lles. are: sufficiently long service life, small overall size, simple manu- re (sleeve bearings), simple and convenient assembly, adjustment and ssembly, etc.

th sliding and rolling friction bearings are used in spindle supports.

Ball and Roller Bearings in Spindle Supports

e high requirements made in respect to the accuracy of spindle rotation ost machine tools are the reason why ball and roller bearings of the e-standard classes of accuracy (П, В, А, С and intermediate classes rding to USSR Std GOST 520-55) are so frequently used in the spindle orts. Moreover, preloaded bearings and bearings with an increased ber of balls or rollers are usually used to reduce the detrimental effect earances and to increase the rigidity of the supports.

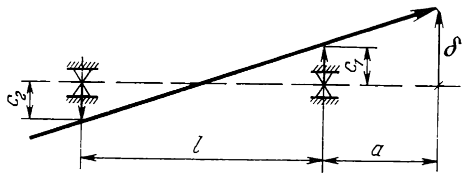


Fig. 55. Diagram for determining the radial runout at the spindle nose

In selecting the class of accuracy of bearings, it is necessary to take into account the substantial increase in their prices with a decrease in the tolerances on radial and axial runout. If the price of a standard (H) accuracy bearing is taken as unity, the prices of the above-standard accuracy bearings will be

H	П	ВП	В	АВ	А	СА	С
1	1.3	1.7	2	3	4	7	10

Hence, in assigning the class of accuracy of the bearings, it is good practice to proceed from the runout of the spindle nose, which is determined from the diagram in Fig. 55, and the relationship

$$\frac{\delta+c_2}{c_1+c_2}=\frac{a+l}{l}$$

or

$$\delta=c_1\left(1+\frac{a}{l}\right)+c_2\frac{a}{l} \tag{124}$$

where c_1 and c_2 are the radial runout values for the front and rear spindle supports, respectively.

Assuming that

$$\delta=\frac{\Delta}{3} \tag{125}$$

where Δ is the tolerance on the radial runout of the spindle nose and that

$$c_1\left(1+\frac{a}{l}\right)=c_2\frac{a}{l}$$

we can write

$$c_1=\frac{\Delta}{6\left(1+\frac{a}{l}\right)} \quad \text{and} \quad c_2=\frac{\Delta}{6\frac{a}{l}} \tag{126}$$

One of the main methods of increasing the accuracy of ball or roller bearings in spindle supports is preloading. This eliminates clearances between the bearing rings and the balls or rollers and, in addition, sets up elastic deformation that improves the total rigidity of the spindle unit. Angular-

contact ball bearings or tapered roller bearings, installed in pairs, are preloaded by adjustments made during assembly, without the need of special devices (Figs. 56 and 57).

Radial ball bearings can be preloaded by axial shift of the inner rings in respect to the outer rings. In practice, this is accomplished by grinding off the end faces of the inner rings (Fig. 58*a*) or by inserting spacers of proper width between the bearing rings (Fig. 58*b*). Relative displacement of the rings can also be achieved by the use of springs (Fig. 58*c*). The last is a more advanced method since it ensures a constant preload which can be more accurately adjusted.

Cylindrical roller bearings are preloaded by expanding the inner ring when the bearing is adjusted axially along a taper journal of the spindle (Fig. 58*d*).

A special design of preloaded rolling friction bearings has found application in the supports of high-speed spindles. These bearings are preloaded when they are being assembled. Thus, the double-row bearing shown in Fig. 59*a* has a split outer ring. When the two halves of this ring are forced together, clearances are eliminated and a preload is produced. Then the half-rings are fixed in this position (for example, by locking rings). The double-row ball bearing shown in Fig. 59*b* is of similar design, but is preloaded during the manufacture and assembly of the bearing by special built-in devices.

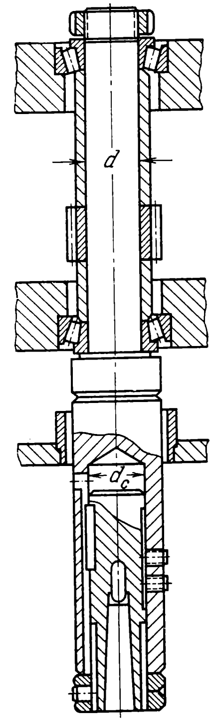


Fig. 56. Spindle of a unit-built multiple-spindle machine tool

Increasing the number of balls or rollers is another means of raising the rigidity of bearings in spindle supports. Standard bearings are available for this purpose with an above-standard number of balls. In recent years, double-row staggered-roller bearings have been widely used in spindle supports (see Vol. 1, Fig. 7, and Fig. 60 in the present volume). In these bearings the number of points of contact around the circumference is doubled.

A very high accuracy of rotation can be attained if compensating elements are provided in the dimension chain made up of the spindle, its support and the housing. The roller bearing rings in Fig. 61, for instance, are machined together with the spindle and with the quill. Narrow tolerances are maintained on the geometrical features (out-of-roundness and taper) of the roller races if there is a possibility of the diameter being changed in

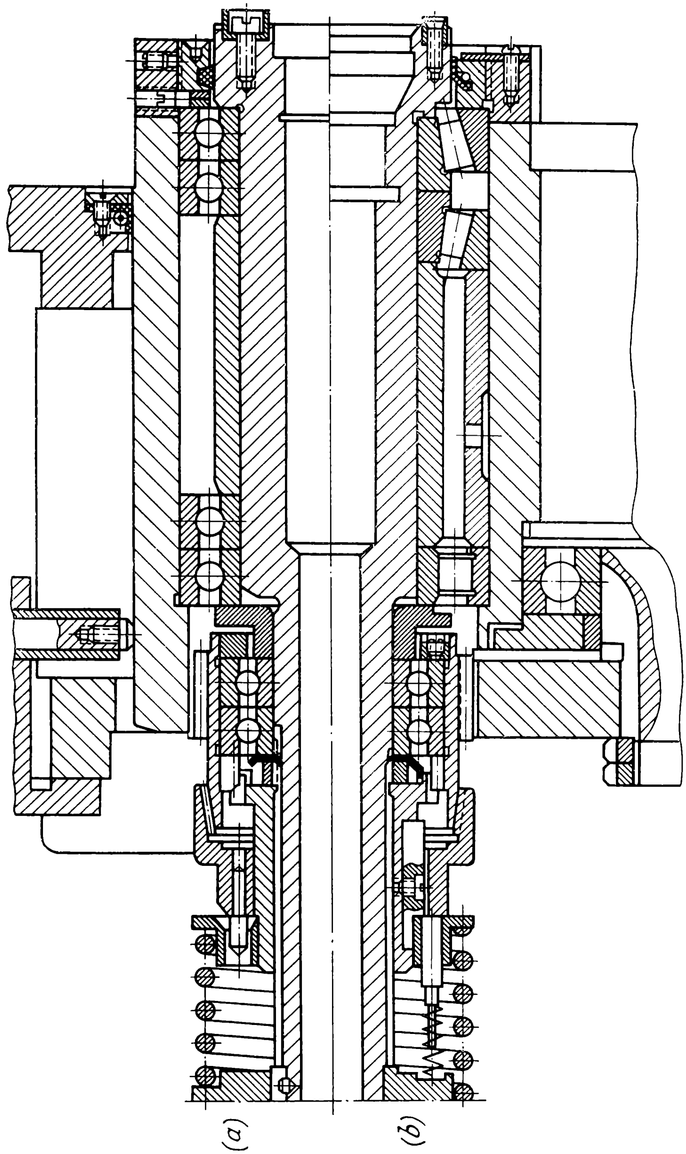


Fig. 57. Spindle unit of a semiautomatic lathe:
(a) before modernization; (b) after modernization

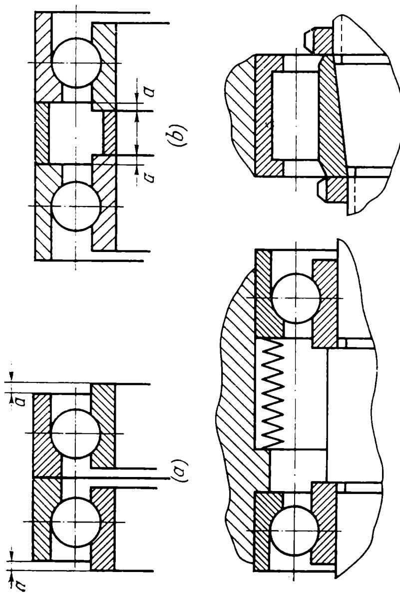


Fig. 58. Methods of preloading bearings

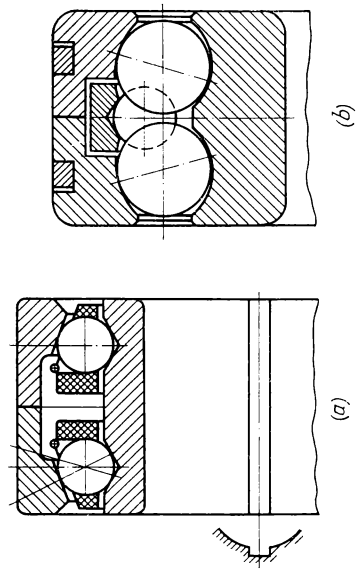


Fig. 59. Double-row ball bearings preloaded in their assembly

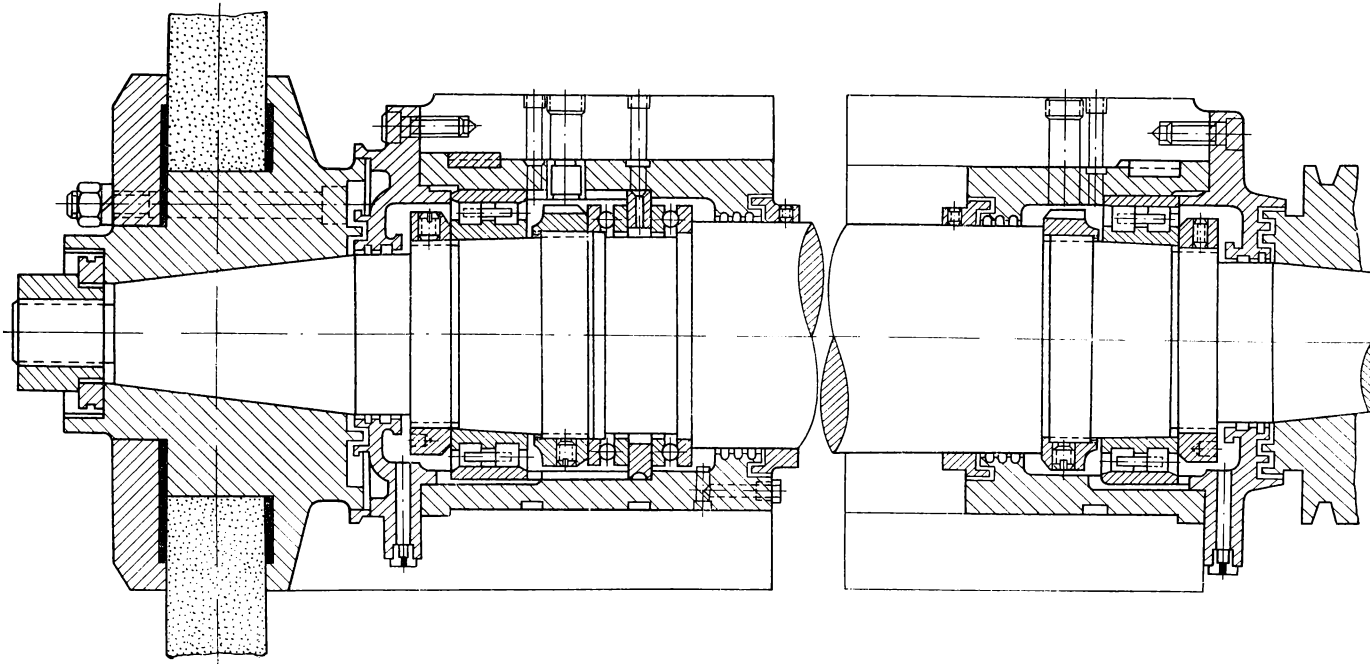


Fig. 60. Spindle unit of a cylindrical grinder

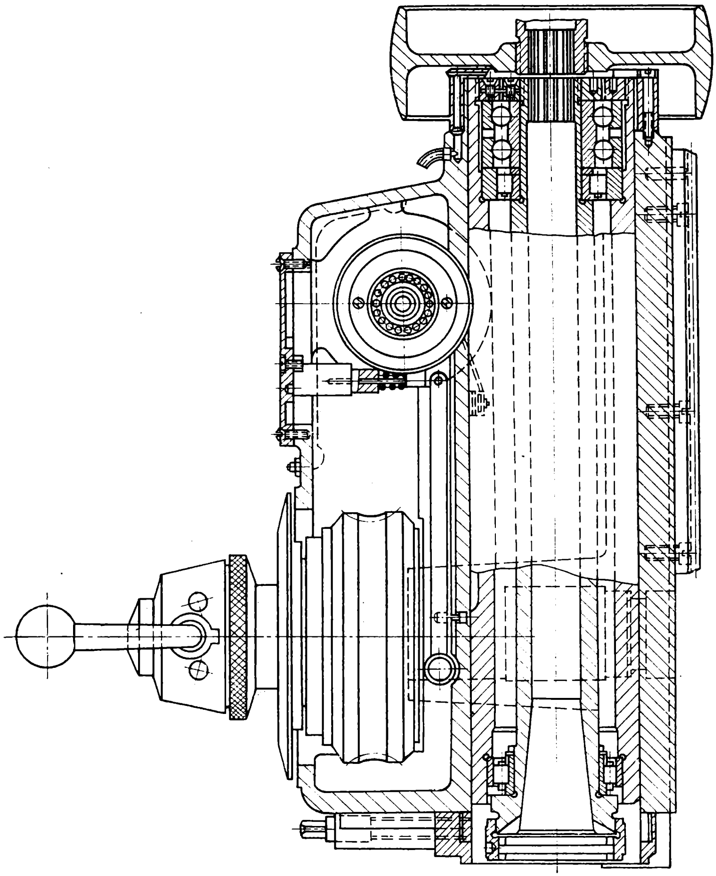


Fig. 61. Spindle unit of a jig borer

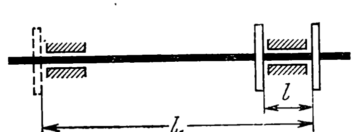


Fig. 62. **Arrangement** of spindle thrust bearings

a fairly wide range. After machining, the diameters of the rings are accurately measured and rollers of the required size are selected, with an accuracy within 1 micron (one half of the maximum difference between the roller diameters).

Rolling friction bearings are used almost exclusively as the thrust bearings for spindles. Two thrust bearings are arranged as close as possible to each other at one of the supports (Fig. 62) to avoid the effects of excessive thermal deformations when the unit heats during operation.

Features of Rolling Friction Bearing Selection for Spindle Supports

The rating life of rolling friction bearings is calculated by the formula

$$C=(Rk_1+mA)(nh)^{0.3}k_bk_h \tag{127}$$

where C = capacity factor

$Rk_1+mA=Q$ = equivalent load, kgf

n = speed, rpm

h = desired life, hrs (in machine tool design, it is usually assumed that $h=5000$ hrs)

k_b and k_h = factors taking into account the nature of the load and which ring (outer or inner) rotates in the support.

Spindle bearing calculations have the following distinctive features:

1. Spindle bearings of general-purpose machine tools may operate at various n and Q values. In this case, the following equivalent load value should be substituted in the preceding formula

$$Q_e=\left(\sum_j \frac{h_j}{h}\,\frac{n_j}{n_e}\,Q_j^{\frac{10}{3}}\right)^{0.3}\text{kgf} \tag{128}$$

where h_j is the life, hrs, of the bearing at the speed n_j and load Q_j.

However, since it is difficult in practice to take into consideration the bearing life at various conditions, the equivalent load is taken approximately as

$$Q_e \cong 0.8Q_{max}$$

and n_e is taken to correspond to the load Q_{max}.

2. Calculations for selecting the bearings for high-speed spindles (internal grinders and certain other machines) should take into account the fact that the speed n rpm may exceed the limiting value n_{lim} rpm indicated in the bearing specifications of the USSR Std (OST). In this case, the factor k_n

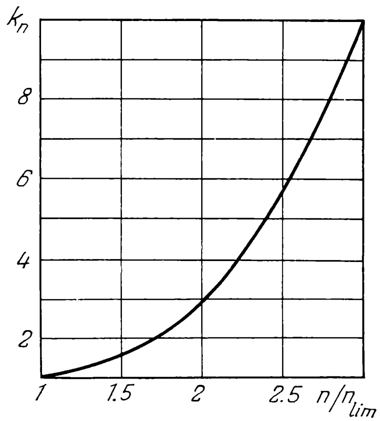


Fig. 63. Correction factor k_n

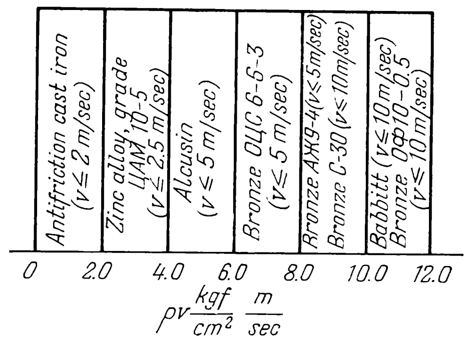


Fig. 64. Bearing material selection chart

is to be introduced in the right-hand side of the main bearing selection formula. Factor k_n can be determined from the diagram in Fig. 63.

3. The preload is usually determined by a formula proposed by D. Reshetov:

$$A_0 = kR \pm 0.5A \tag{129}$$

where $k = 0.5$ to 0.6 for radial ball bearings with above-standard clearances and spindle bearings, and $k = 0.65$ to 0.8 for angular-contact ball bearings. The plus sign in the formula is used when the acting thrust load reduces the preload and the minus sign when it increases the preload.

Materials Used for Sliding Friction Spindle Bearings

Factors to be taken into account in selecting the materials for sliding friction bearings of spindles are wear-resistance, heat conductivity, coefficient of friction, coefficient of linear expansion and, in some cases, certain other properties of antifriction alloys.

A tentative selection of the material can be made on the basis of the peripheral velocity v and the specific pressure p (Fig. 64).

Cast irons possess poor running-in, or break-in, properties and therefore the surfaces of a cast iron bearing sleeve and the hardened journal of the steel spindle should be carefully finished. The spindle should be sufficiently stiff to avoid edge bearing contact pressure.

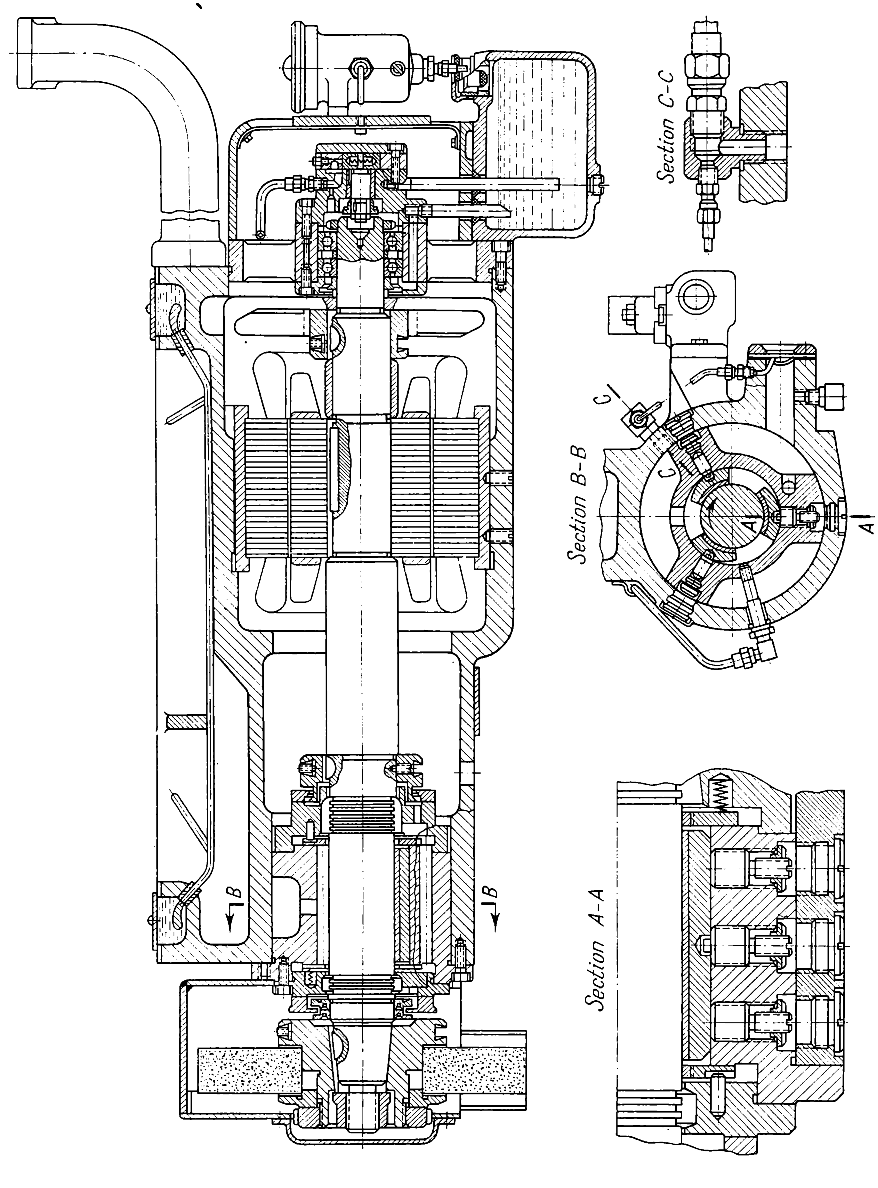


Fig. 65. Spindle unit of a surface grinder

At low peripheral velocities (tenths and hundredths of m per sec), cast iron bearings are capable of withstanding pressures up to 200 or 300 kgf per sq cm.

Bronzes, due to their relatively high cost, are used as bimetal bushings or sleeves. The steel or cast iron back is lined with a thin layer (~ 1 mm, after being machined) of bronze (Fig. 65). If solid bronze bearings are replaced by bimetal sleeves, the consumption of nonferrous metals is frequently reduced by 75 to 80 per cent, and sometimes by 90 per cent, the cost of the bearing is reduced by 60 to 70 per cent, while the service life is increased, especially if the lining is thin. Bimetal bearings are usually lined by centrifugal-casting techniques in machine tool plants.

Tin bronzes should be used only in cases where their necessity is justified by calculations or experimental data.

Babbitts are used in the form of bimetal bushings in large bearings. Babbitt bearings have good running-in properties, due to which they provide excellent service in operation with an unhardened journal.

Constructions of Sliding Friction Bearings

Nonadjustable bearings are not frequently used as spindle supports, and then only in cases when the operating conditions are such that a practically complete absence of wear can be expected over a long period of service (low-speed and lightly loaded spindles of microfinishing machines, etc). The dimensions of solid nonadjustable bearing sleeves of bronze or cast iron have been standardized.

More widely used as spindle supports are sliding friction bearings whose design incorporates means for periodical (manual) or continuous (automatic) clearance adjustment. Automatic adjustments are accomplished by either spring or hydraulic action.

Bearings with radial clearance adjustment. The sleeve in these bearings consists of two, three and sometimes more members. Some of the members are stationary while the rest are movable in the radial direction, this movement being used to adjust the clearance between the spindle journal and the bearing (Fig. 66). The main advantage of this type of bearing is the convenience offered in assembling and disassembling the spindle unit. This feature has found these bearings wide application as the spindle supports of heavy machine tools.

Bearings with axial clearance adjustment. The bearing sleeve has a through slot along its full length (Fig. 67*a*) or is made solid (Fig. 67*b*). Clearance is adjusted by axial movement of the sleeve. In making adjustments on the first type of these bearings (Fig. 67*a*), the cylindrical shape of the sleeve bore

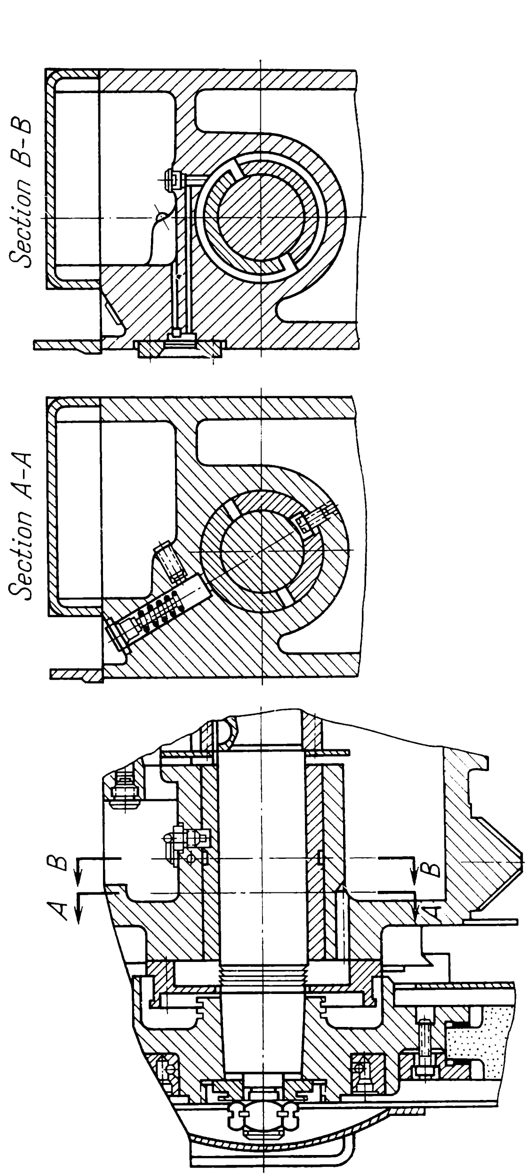


Fig 66. Front spindle support of a cylindrical grinder

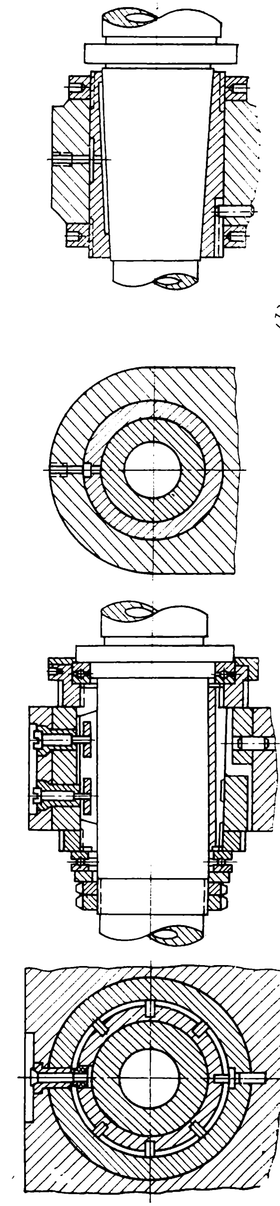


Fig. 67. Sliding friction bearings with axial clearance adjustment

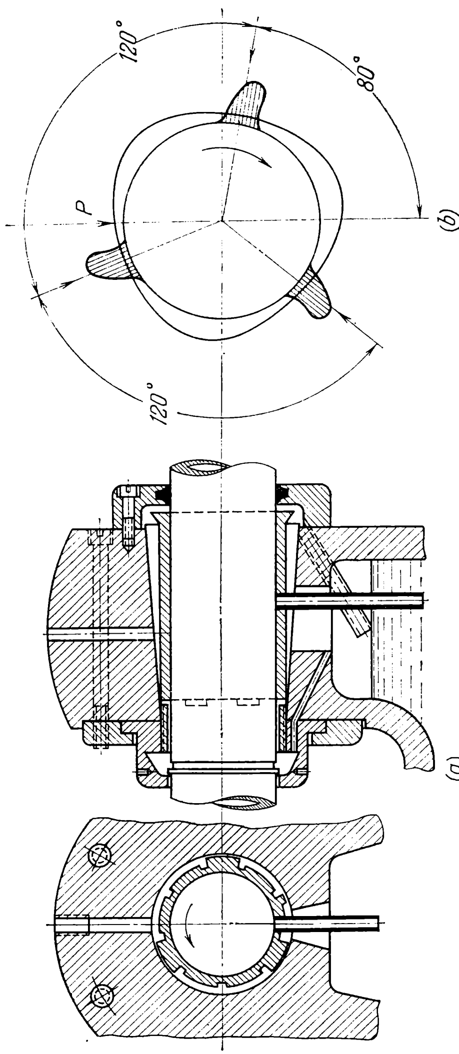


Fig. 68. Multiple-wedge bearing with a noncircular bore

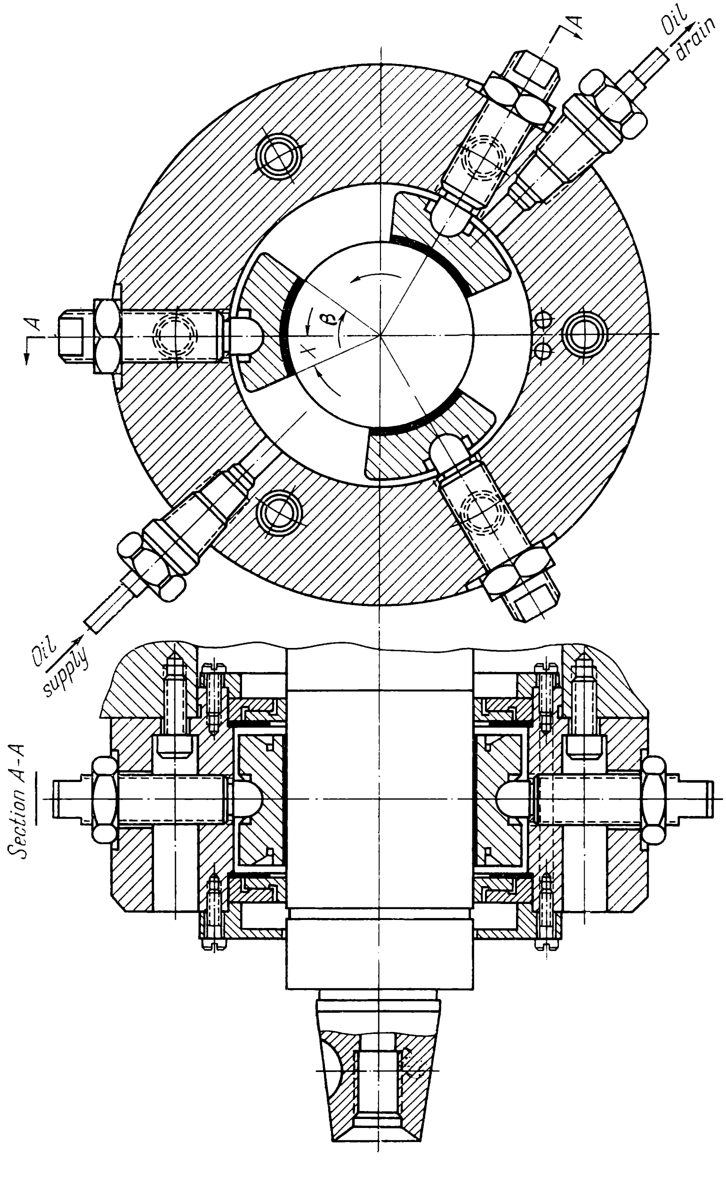


Fig. 69. Self-aligning tilting-pad bearing

is distorted to some extent. A drawback of the second type is the disarrangement of the adjusted clearance upon axial displacement of the spindle.

Multiple-surface (multiple-wedge) bearings ensure high accuracy of rotation because the spindle is centred by hydrodynamic pressure developed in oil wedges at several zones around the circumference. These wedge-shaped oil pockets are produced in this type of bearing either by nonuniform deformation of the sleeve (Fig. 68*a* and *b*) or by using a bearing made up of several self-aligning segments (Fig. 69) spaced equally around the circumference. This second type is also known as a tilting-pad bearing.

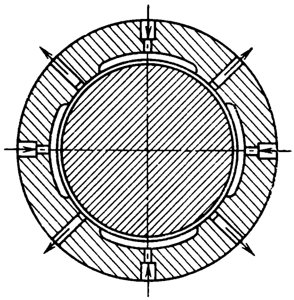


Fig. 70. Hydrostatic (externally pressurized) bearing

Hydrostatic (externally pressurized) bearings have provisions for supplying oil at considerable pressure to several pockets in the bearing (Fig. 70). The oil flows out through the clearances between a shoulder on the spindle and the end of the bearing. Hydrostatic bearings can operate under fluid friction conditions even at the slowest speeds of rotation.

Air bearings can operate with aerodynamic pressure at high speeds of rotation, or they are designed as aerostatic supports with large surplus pressure of the air supply. Features of air bearings are their lower rigidity as compared to hydraulic bearings and lower friction losses. Both factors are due to the fact that the viscosity of air is only $\frac{1}{2000}$ that of Industrial oil 20.

CHAPTER 6

MECHANISMS FOR RECTILINEAR MOTION

6-1. Methods for Producing Rectilinear Motion in Machine Tools

Rectilinear motion is produced in machine tool drives by one of the following principal methods:

1. By the use of a hydraulic device based on a piston and cylinder as the kinematic pair that powers the rectilinear motion. A hydraulic drive of this type has many advantageous features, thanks to which it is so widely employed in various machine tools as a main drive, as well as a feed drive or a drive for auxiliary movements. All aspects of hydraulic drives for machine tools are considered in Part Four (Vol. 2).
2. By using electromagnetic devices of the solenoid type. The limited stroke of these devices and their operation with impacts permit them to be used only in the drives of control systems as auxiliary devices.
3. By means of mechanisms that convert rotary motion into rectilinear motion. These include such kinematic pairs as the pinion and rack, worm and rack, screw and nut, etc.

The first two classes of methods listed above are taken up in Part Four (Vol. 2) dealing with hydraulic systems and equipment, and in textbooks on electrical circuits of machine tools. Hence, the following will concern only the main features of the third group of mechanisms as applied to the drives of metal-cutting machine tools.

6-2. Rack and Pinion Drives

The following features of rack and pinion drives are the most essential to their application as machine tool drives:

1. They have a large transmission ratio—upon each revolution of the pinion, the rack travels a distance equal to the length of the pinion pitch circle. Consequently, they can be conveniently used in main drives and in the drives of various auxiliary motions.
2. The transmission ratio is not uniform because the errors in the gearing affect the velocity of rack motion. It is especially difficult to maintain uniform slow travel in the feed drive of a high-precision machine tool with a pinion and rack.

3. The high efficiency of this pair enables it to be employed in drives for transmitting considerable power, for instance in the main drive of planers and slotters.
4. The lack of self-braking in a rack and pinion drive presents difficulties when it is used for vertical positioning movements. On the other hand, this drive can be used in parallel with another type of drive—a lead screw—because it has no self-braking properties.
5. Both racks and pinions are easy to manufacture and have a relatively low cost.

Materials Used for Pinions and Racks

Large rack pinions and racks of planers are made in the Soviet Union of grey cast iron, grade СЧ 21-40, СЧ 28-48 or СЧ 38-60, or of steel 45 which undergoes structural improvement and tempering to a hardness Bhn 230-260.

In the design of feed mechanisms, the diameter of the pinion and, consequently, the pitch of the traversing element is made as small as possible to reduce the torque on the traversing shaft (pinion shaft in this case) and to shorten the reduction train of the feed drive. With this in view, the pinion is made of alloy steel, and the rack of alloy steel or structural steel 45. Heat treatment is assigned with the aim of increasing, not only the beam strength of the teeth, but the surface endurance limit of the material (bearing strength) as well. Surface contact pressures frequently deform the teeth of unhardened racks. Induction hardening of the rack teeth reduces the tendency of the rack to distortion in heat treatment.

Racks from 1000 to 1200 mm long are cut in rack milling or plain horizontal milling machines equipped with a fixture for indexing the table one pitch in cutting each tooth. Long racks are made of two or more sections. The rack is located with dowel pins and secured to the corresponding part of the machine with screws.

Racks for feeding drill press spindles are sometimes cut directly on the spindle quill.

Design of Rack and Pinion Drives

Rack and pinion transmissions in the main drive of a machine tool are subject to considerable speeds and loads. They are designed on the basis of calculations used for toothed gearing. These calculation methods are given in the Machine Design course.

In feed mechanisms there is no necessity to check the wear strength of rack and pinion drives; it is sufficient to check the beam and bearing strengths

of the teeth. The following formula is used to test the bearing strength of the tooth surfaces

$$Q = 1.4q^2 \frac{zmb \sin 2\alpha}{E} \text{ kgf} \tag{130}$$

where Q = permissible peripheral (tangential) force acting on the rack pinion, equal to the feeding force, kgf
q = maximum permissible bearing stress in contact of the rack and pinion on the pitch circle, kgf per sq cm
z = number of teeth on the rack pinion
α = pressure angle of the gearing
m = module, mm
b = face width of the pinion, mm
E = modulus of elasticity, kgf per sq mm.

The permissible bearing stress is taken equal to $q < 3\sigma_T$, where σ_T is the yield stress of the material.

6-3. Worm and Rack Drives

Unlike rack and pinion drives, a worm and rack drive can be used to obtain low transmission ratios. Moreover, much smoother motion is produced. On the other hand, worm and rack drives are more complicated in manufacture than rack and pinion drives, and have a lower efficiency than ordinary worm gearing.

Materials Used and Design Features

The materials used to make the worm and rack should have good antifriction properties because much sliding motion is involved in the operation of these drives. The worm is usually made of casehardening steel 15X or 20X which is then carburized and hardened. The rack is made of antifriction cast iron. In the most critical applications, a bimetal rack may be used in which the teeth are cut in a layer of bronze. There have been cases in which a bronze worm was used. This led to intensive wear of the worm. However, a worn worm can be more easily and cheaply replaced than a worm rack whose manufacture requires the use of special cutting tools and equipment.

The following types of worm and rack drives are employed in machine tools:

1. Worm and gear rack drive. This arrangement has point contact between the worm thread and the rack teeth, and is used mostly for the drives of auxiliary motions.

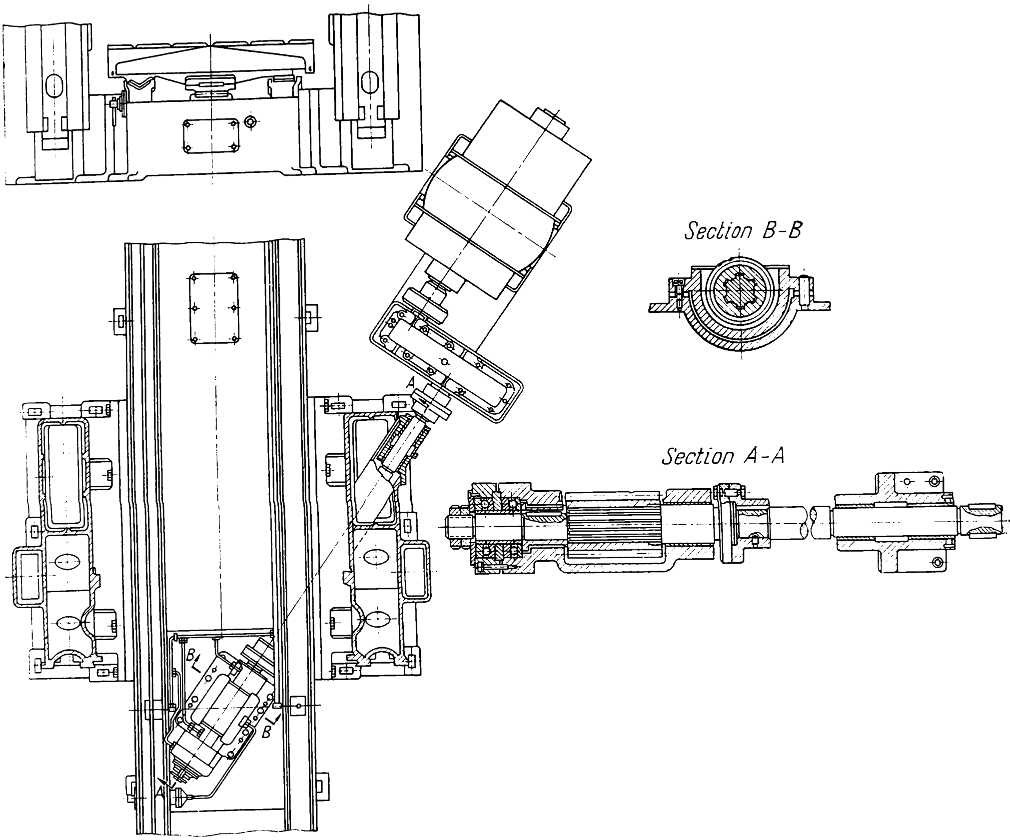


Fig. 71. Worm and rack drive of a planer table

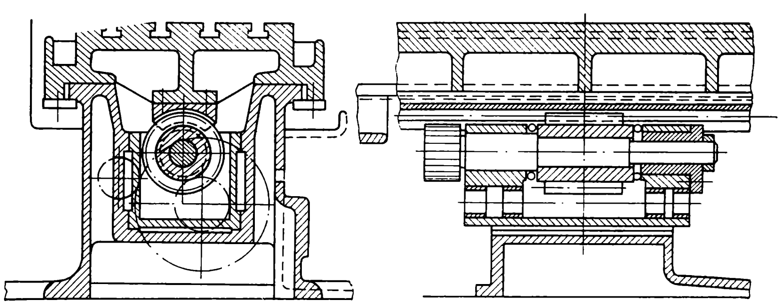


Fig. 72. Worm and rack table-feed drive in a planer-type milling machine

2. Worm and nut-type rack drive with the axis of the worm arranged at an angle to the rack axis (Fig. 71). The teeth of the rack resemble those of a worm wheel and the type of engagement is the same as in ordinary worm gearing.

3. Worm and nut-type rack with a parallel arrangement of the worm and rack axes (Fig. 72). The type of engagement is like that of a short screw and an incomplete (partly enveloping) nut. In a construction based on this arrangement, the outside diameter of the gear in the worm drive must be less than the root diameter of the worm. This condition is avoided in some cases by using a worm on which gear teeth have been cut in addition to the worm thread.

6-4. Lead Screw and Nut Drives

The extensive application of mating screws and nuts in rectilinear motion drives of machine tools is due to the following features:

1. The low transmission ratio, when single-start thread is used, enables slow motions to be obtained in the feed drive.

2. An exceptionally smooth and highly accurate motion is produced due to the strictly constant transmission ratio of a power screw and nut. The degree of accuracy and the smoothness of the motion are determined primarily by the accuracy to which the screw and nut have been manufactured.

3. The low efficiency of ordinary (sliding friction) power screws and nuts hinders their use in main drives, but is not such an essential drawback for feed and auxiliary motion drives. The efficiency of ball-bearing screws will be considered further on.

4. The self-braking capacity of ordinary screws and nuts facilitates their application for positioning and vertical movements.

Manufacturing Specifications

Standard ТУД 22-2 of the Soviet machine tool industry established five accuracy classes for lead screws. They are: 0 (the most accurate), 1st, 2nd, 3rd and 4th. The standard stipulates the maximum permissible: (a) pitch error (between adjacent threads and maximum accumulated pitch error for various lengths of thread), (b) half angle of thread error, (c) out-of-roundness of the thread at the pitch diameter, and (d) runout at the major diameter.

In respect to lead screw nuts, the standard establishes plus tolerances for the pitch diameter.

Accurate operation and a reduction in wear of a power screw and nut additionally require: (a) that the runout of the lead screw thread in reference

to the screw journals be limited, (b) that the axis of the lead screw bearings be parallel to the corresponding ways, (c) that the lead screw has no axial runout in its rotation, and (d) that the lead screw and its nut be strictly coaxial.

Materials Used for Lead Screws and Nuts

The materials for making lead screws and nuts are selected in accordance with the purpose of the screw, its class of accuracy (see below) and required heat treatment. Machine tool standard ТУД 22-2 recommends: (a) carbon tool steel. grade У10 or У12 for 0 class accuracy lead screws used in precision machine tools (for instance, jig borers), (b) steel ХВГ or ХГ, hardened to 50-56R_C, or steel 65Г, hardened to 35-45R_C, for 1st class accuracy lead screws that require high hardness after heat treatment (for example, thread grinders), and (c) steel 45 or 50 of standard composition, steel 45 with a 0.15 to 0.5 per cent addition of lead, or free-cutting steel А40Г containing 1.20 to 1.55 per cent manganese for lead screws of 2nd class accuracy (for standard accuracy engine lathes), of 3rd class accuracy (for milling machines and planers), and of 4th class accuracy (for positioning movements), which are not to be heat treated to a high hardness. Smooth surfaces, with no scoring, are obtained in thread cutting if the lead screws are made of either of the last two types of steel (steel with a lead addition and free-cutting steel). The steel undergoes a special heat treatment that reduces the deformation of the screw blank in subsequent machining operations.

Lead screw nuts are usually made of tin bronze. Standard ТУД 22-2 recommends tin bronze, grade Бр. ОФ 10-0.5 or Бр. ОЦС 6-6-3, for the nuts of lead screws of the 0, 1st and 2nd accuracy classes. The nuts of 3rd and 4th accuracy class lead screws can also be made of antifriction cast iron.

Constructions of Lead Screw and Nut Drives

Thread forms. In Soviet practice, lead screws are usually made with standard trapezoidal thread (USSR Std GOST 9484-60) having a 30° angle of thread. In comparison to square threads, trapezoidal threads possess the following advantages: (a) trapezoidal threads can be milled and ground without distortion of their form, and (b) they permit easier closing of half-nuts (full closing of the half-nuts is impossible with square threads if the half-nuts fully envelope the screw). One drawback of trapezoidal threads is that pitch errors result from radial runout of the lead screw during the thread-cutting operation. This has been overcome in making the lead screws of high-precision thread-cutting machines by employing square threads, or trapezoidal threads with a smaller angle of thread (10° to 15°).

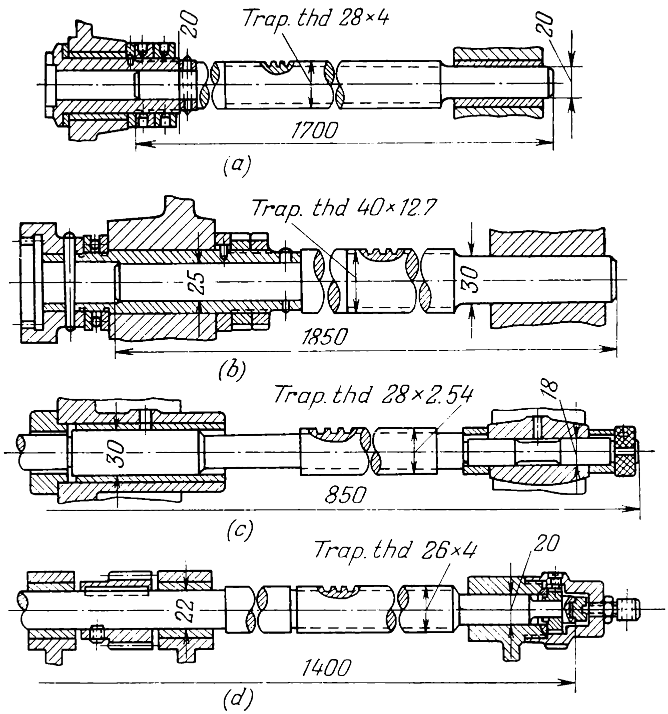


Fig. 73. Engine lathe lead screw units:
(*a*) and (*b*) standard accuracy; (*c*) and (*d*)—above-standard accuracy

Lead screw bearings. Bearings of lead screws should be designed so that they do not allow excessive axial and radial runout of the screw which may lead to pitch errors in the thread being cut.

Thrust bearings should be arranged so that heating of the screw does not result in dangerous thermal stresses and deformation. For this reason, lead screws are fixed axially, in most cases, only in one support. Only heavily loaded lead screws, subject to tensile stresses, are fixed axially in both supports.

Sliding friction bearings, designed as bronze or antifriction cast iron bushings, are used for lead screws more frequently than ball or roller bearings since they have the following advantages: it is easier to attain a small runout, bushings have smaller overall size (an important factor if the slide member is to pass over the bearing) and the construction of the unit is simpler if bushings are used (Fig. 73).

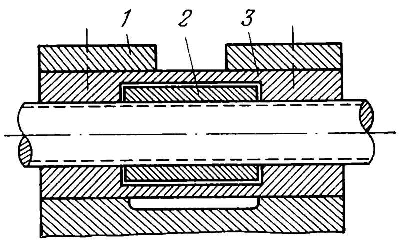


Fig. 74. Additional support of the lead screw in a high-speed lathe

Ball and roller bearings are used, in the main, for heavily loaded lead screws of medium accuracy.

Lead screws are fixed in the axial direction either by ball thrust bearings of above-standard accuracy or by sliding thrust bearings (Fig. 73*a* and *c*). The latter are preferable in precision machine tools. To reduce axial runout, the diameters of the bearing surfaces are made as small as possible and self-aligning spherical thrust rings are employed (Fig. 73*d*).

Constructions that reduce bending deformation of lead screws. Various methods are applied to reduce the deflection of lead screws. These include:

(a) The rigidity of the screw bearings is raised by using bushings with a higher length-to-bore diameter ratio, i.e., length-to-diameter ratio of the screw journal. This enables a single bearing to be used for short lead screws, the second support being the nut. Screws of medium length require two bearings in any case.

(b) Additional supports are provided for long lead screws. If the screw is not too heavy, the support is designed as a sleeve of sufficient length whose bore is an exact fit on the major diameter of the screw thread. This sleeve travels together with the nut. In the construction shown in Fig. 74, sleeve *3* is fitted and secured in saddle *1* and has a recess to accommodate nut *2*.

(c) The deflection of long heavy lead screws is reduced by the provision of a hinged support which is pushed away by the saddle as it travels by, or of supports which only partly envelop the screw (Fig. 75). In the second case, the nut is also designed for partly enveloping the screw. This is an essential drawback of such a construction because the eccentric application of the feeding force develops a moment that tends to bend the screw.

The lead screw is commonly arranged between the ways in the middle plane of the bed (or base) to reduce the moment which tends to swivel the table or saddle in a horizontal plane. Lead screws are thus arranged in high-

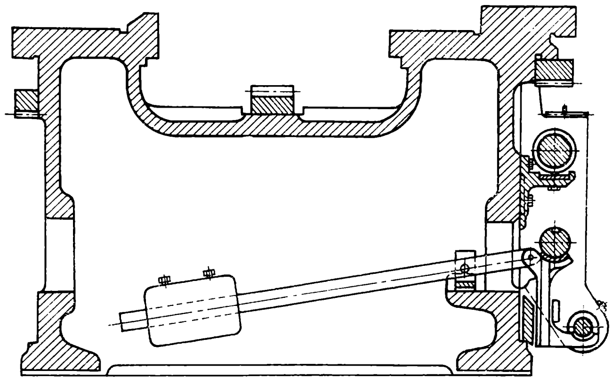


Fig. 75. Hinged support of a long lead screw

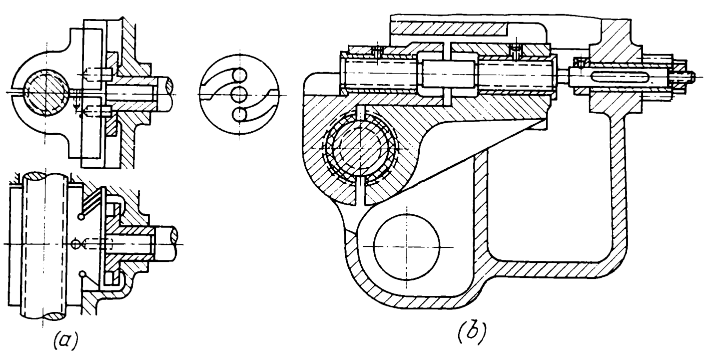


Fig. 76. Designs of lead screw half-nuts

accuracy thread-cutting lathes to increase the machining accuracy, and in heavy lathes, planers, milling machines, etc., to improve the working conditions of the screw and ways.

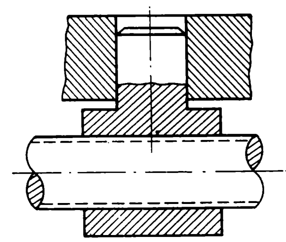


Fig. 77. Self-aligning solid nut

Constructions of lead screw nuts. Lead screws for working feeds usually have single- or two-start threads of small pitch (with a correspondingly small helix angle) and are therefore self-locking. Hence, if the design of the machine tool incorporates some other traversing device (for example, the rack and pinion feed drive in engine lathes), in addition to the lead screw, the nut of the lead screw should be of the split type (as half-nuts). Half-nuts and their opening and closing mechanism are shown in Fig. 76*a* and *b*. If, on the other hand, the lead screw is the only traversing device, a solid nut is used. The simplest construction of such a nut is illustrated in Fig. 77. Here the axes of the nut and screw can be aligned in one plane but not in the other. This is a drawback of this construction.

Lead screw nuts are frequently made of two sections to make it possible to adjust the clearance in the thread and to eliminate backlash. One section is fixed stationary on the slide while the other can be adjusted axially by means of a wedge (Fig. 78), set nut (Fig. 79), spring (for small loads on the screw, as in grinders, Fig. 80*a*) or hydraulic pressure (Fig. 80*b*). A design of the last type is used in milling machines adapted for climb-cut milling. When the same machine is used for conventional milling, or during rapid traverse movements, a valve connects the cylinders at both ends of the nut, thereby removing the load on the nut.

Lead compensating (correction) devices. Machine tools designed for cutting precise threads have a device for automatically compensating pitch errors of the lead screw during operation (Fig. 81). The principle of this device may be based either on: (a) supplementary axial displacement of the lead screw (Fig. 81*a*) or, more often, (b) supplementary rotation of the clasp nut (Fig. 81*b*). These devices were considered in detail in Chapters 18 and 19 of Vol. 1.

Lead screw pitch errors, caused by temperature deformations in the process of operation and having a regular nature, can be compensated by straight, swivelling correction bars, set at the proper angle.

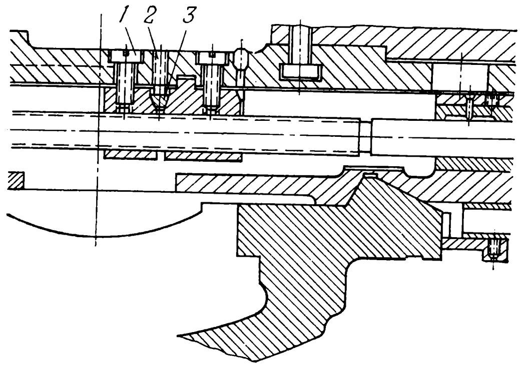


Fig. 78. Two-section nut with a wedge for periodical backlash adjustment: *1*—clamping screws; *2*—adjusting screw; *3*—wedge

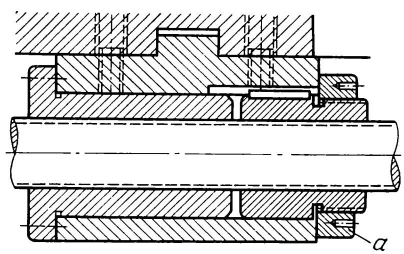


Fig. 79. Backlash adjustment by means of set nut *a*

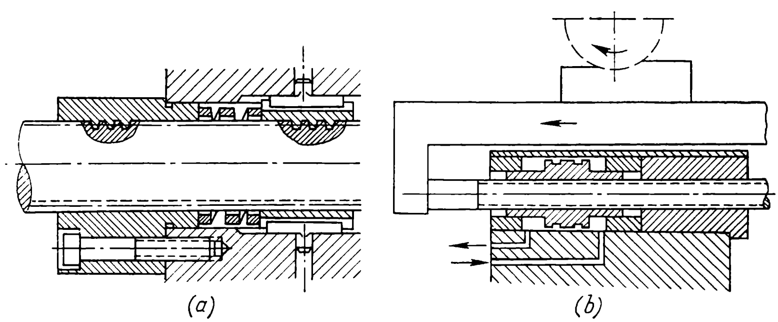


Fig. 80. Nuts with automatic backlash elimination

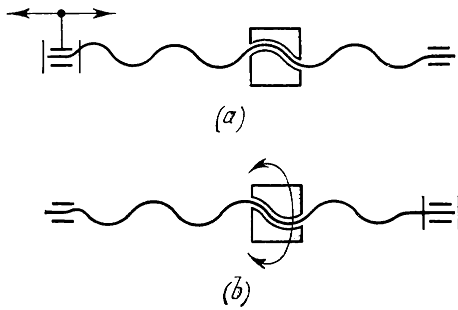


Fig. 81. Diagrams of lead compensating devices:
(a) with supplementary axial displacement of the lead screw; (b) with supplementary rotation of the nut

Design of Screw and Nut Drive Mechanisms

The dimensions of the lead screw and nut are determined on the basis of wear resistance, strength and rigidity of the construction and of the buckling stability of the lead screw.

1. *Wear-resistance calculations* are carried out by determining the average specific pressure on the working surfaces of the thread by means of the formula

$$p = \frac{100Q}{\pi d_p t_2 \frac{Lz}{s}} = \frac{100Qs}{\pi d_p t_2 Lz} \text{ kgf per sq cm} \tag{131}$$

where Q = maximum traversing (feed) force, kgf
s = thread lead, mm
t_2 = height of thread engagement, mm
L = length of the nut, mm
z = number of starts
d_p = pitch diameter of the thread, mm.

Denoting $\frac{L}{d_p}$ by λ' and substituting, we obtain

$$p = \frac{100Qs}{\pi \lambda' d_p^2 t_2 z} \text{ kgf per sq cm}$$

from which

$$d_p \cong 5.6 \sqrt{\frac{Qs}{\lambda' z t_2 p}} \text{ mm} \tag{132}$$

In standard trapezoidal threads $t_2 = 0.5 \frac{s}{z}$; substituting for t_2 in the last equation we can write

$$d_p \cong 8 \sqrt{\frac{Q}{\lambda' p}} \text{ mm} \tag{133}$$

The ratio $\lambda' = \frac{L}{d_p}$ is taken in the range from 1.5 to 4; for half-nuts (clasp nuts), $\lambda' \cong 3$.

The permissible average specific pressure may have the following values: (a) $p = 30$ kgf per sq cm for a steel lead screw and a bronze nut intended for precise feeds (thread-cutting machines, engine lathes and thread-milling machines) and (b) $p = 120$ kgf per sq cm for other critical lead screws (milling machines) mating with a bronze nut, and $p = 80$ kgf per sq cm for the same lead screws mating with a cast iron nut.

2. *Strength calculations for lead screws.* A lead screw is subject to combined tensile (or compressive) and torsional stresses, and strength calculations are based on the equivalent stress which is

$$\sigma_e = \sqrt{\sigma^2 + 4\tau^2} = \sqrt{\left(\frac{Q}{F}\right)^2 + 4\left(\frac{M_t}{W_p}\right)^2} \text{ kgf per sq mm} \tag{134}$$

where $F = \frac{\pi d_1^2}{4}$ = area of the minor-diameter cross section, sq mm
M_t = torque transmitted by the screw, kgf-mm
$W_p = \frac{\pi d_1^3}{16} = F\frac{d_1}{4}$ = sectional modulus of torsion, cu mm (the effect of the threads on W_p is neglected).

After substituting, we can write

$$\sigma_e = \frac{1}{F}\sqrt{Q^2 + \left(\frac{8M_t}{d_1}\right)^2} \text{ kgf per sq mm} \tag{135}$$

The torque transmitted by the screw [see equation (102)] is

$$M_t = \frac{Qs}{2\pi\eta} \text{ kgf-mm} \tag{136}$$

where η is the efficiency of the screw and nut pair, determined by the formula

$$\eta = \frac{\tan\beta}{\tan(\beta + \rho)} \tag{137}$$

where β = helix angle of the thread at the pitch diameter
$\rho = 6°$ to $8°$ = angle of friction of the thread
Q and s are the same as in equation (131).

The permissible equivalent stress is assigned in accordance with the yield strength σ_T of the screw material so that

$$\sigma_T \geqslant (3 \text{ to } 3.5)\,\sigma_e \text{ kgf per sq mm} \tag{138}$$

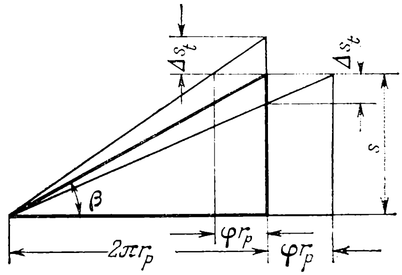


Fig. 82. Diagram for lead screw rigidity calculations

3. *Rigidity calculations.* The change in the thread pitch, due to compression or tension of the screw from the traversing (feeding) force Q, is

$$\Delta s_Q = \pm \frac{Qs}{EF} \text{ mm} \tag{139}$$

where E is Young's modulus of elasticity, kgf per sq mm; s and F are the same as above.

It can be seen in Fig. 82 (development of one turn of thread on a plane) that the change in the thread pitch, due to the twist of the lead screw from the torque M_t, is

$$\Delta s_t = \pm s \frac{\varphi}{2\pi \mp \varphi} \simeq \pm s \frac{\varphi}{2\pi} \text{ mm} \tag{140}$$

The angle of twist of a lead screw over the length of one pitch is

$$\varphi = \frac{M_t s}{E_s J} \text{ rad} \tag{141}$$

hence

$$\Delta s_t = \pm \frac{M_t s^2}{2\pi E_s J} \text{ mm} \tag{142}$$

where E_s = modulus of elasticity in shear, kgf per sq mm
J = polar moment of inertia of the screw cross section, mm^4.

Analyzing equations (139) and (142), we find that $\Delta s_t \ll \Delta s_Q$, i.e., variation in pitch is primarily due to axial deformation. Thus, in determining the rigidity of a lead screw, changes in pitch due to torsion can be neglected.

The permissible increase or decrease in the thread pitch should be assigned on the basis of the pitch tolerance for a lead screw of the corresponding accuracy class.

4. *Buckling stability calculations.* If the lead screw operates under a compressive load and its length is considerable in comparison with its diameter,

it should be checked for buckling stability as a slender column loaded by the centrally applied compressive force Q—the maximum traversing force.

The critical traversing (feeding) force is

$$Q_{cr} = \frac{\pi^2 E I_{min}}{(\nu l)^2} \text{ kgf} \tag{143}$$

where E = Young's modulus of elasticity, kgf per sq mm
I_{min} = minimum moment of inertia of the cross sections, mm^4
νl = reduced buckling length, mm.

The margin of buckling stability n_{bs} is determined from the formula

$$n_{bs} = \frac{Q_{cr}}{Q} = \frac{\pi^2 E I_{min}}{\nu^2 Q l^2} \tag{144}$$

The length factor ν can be taken as follows: for two fixed ends $\nu = \frac{1}{2}$; for one fixed and one pinned end $\nu = \frac{1}{\sqrt{2}}$ and for two pinned ends $\nu = 1$. Machine tool industry standard H48-62, developed by D. Reshetov and G. Levit (ENIMS), recommends that the end conditions of screws in their bearings be established in accordance with $\lambda_b' = \frac{l_b}{d_b}$, where l_b is the length of the bearing and d_b is its bore diameter. This ratio is used as follows: if $\lambda_b' \leqslant 1.5$, the bearing is considered as a pinned end; if $\lambda_b' \geqslant 3$, the screw can be said to be perfectly fixed in the bearing; and if $1.5 < \lambda_b' < 3$, the screw is imperfectly fixed in the bearing. In this last case, if the other end is perfectly fixed, then $\nu = \frac{1}{\sqrt{2.8}}$.

The end conditions for a solid nut are considered to be the same as for a bearing, in accordance with the ratio of the nut length to the thread pitch diameter. Half-nuts are treated as an imperfectly fixed end.

The margin of buckling stability is taken in the range $n_{bs} = 2.5$ to 4. Larger values are taken if the screw is subject to transverse forces developed by the drive.

The buckling stability is checked only for long lead screws in which $\nu l > (7.5 \text{ to } 10)\, d_1$.

Rolling Friction Screws and Nuts

Various types of screws and nuts, operating with rolling friction, were developed to eliminate the detrimental effects of sliding in threads and consequent wear. Certain of these types are being used more and more in machine tools.

In addition to the low friction losses and the high efficiency (Fig. 83), an important advantage of rolling friction screws and nuts is that they can be preloaded so as to completely eliminate backlash. Backlash is extremely undesirable in cases of alternating axial loads and reversible precise motions (as in the drives of numerically controlled machine tools).

Rolling friction can be substituted for sliding friction in screw and nut pairs either by using rollers, rotating freely on their axles, instead of nuts (Fig. 84), or by employing balls (and sometimes rollers) running along in the thread between the screw and nut on a recirculating principle with a return passage (Fig. 85). Because of difficulties encountered in manufacture and assembly, constructions embodying rollers (Fig. 84) have not found as wide application in machine tools as ball-bearing screws.

The thread of ball-bearing screws and nuts is usually of half-round (Fig. 86*a*) or of ogive (Gothic arch) form as in Fig. 86*b*. In both cases, the small difference in the curvature of the balls and the raceway (thread) increases the contact area and thereby reduces the contact stresses.

As a rule, ball-bearing screw and nut pairs incorporate devices for backlash elimination and preloading (Fig. 87).

The design of ball-bearing screws and nuts includes:

1. *Static strength calculations* in accordance with permissible contact stresses. In standard screw and nut pairs, the maximum permissible static load on one ball is

$$P_{st} = 2d_1^2 \text{ kgf} \qquad (145)$$

where d_1 is the ball diameter, mm.

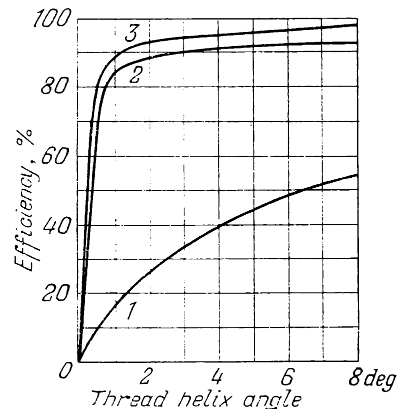


Fig. 83. Friction losses in sliding and rolling friction screws and nuts: *1*—sliding friction screw and nut; *2*—ball-bearing screw with rolled thread; *3*—ball-bearing screw with ground thread

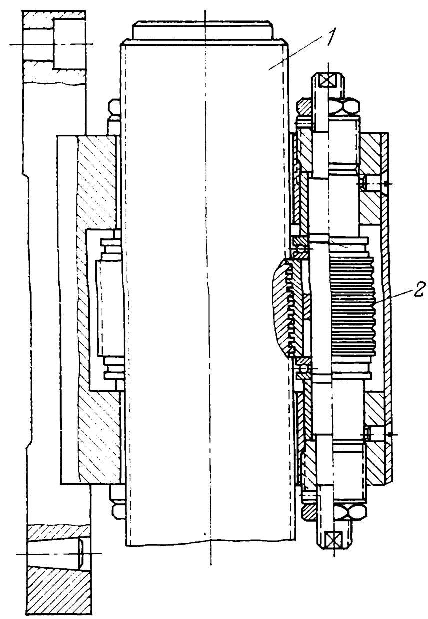


Fig. 84. Screw with a roller-type nut: *1*—screw; *2*—roller

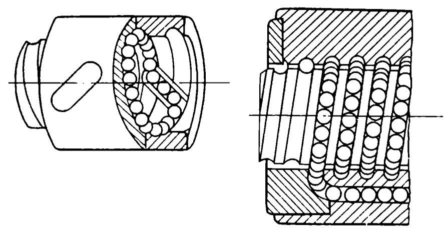


Fig. 85. Ball-bearing screw and nut

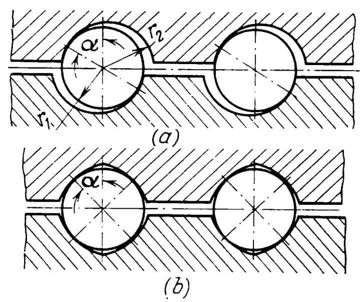


Fig. 86. Thread forms for ball-bearing screws and nuts:
(*a*) half-round; (*b*) ogive (Gothic arc)

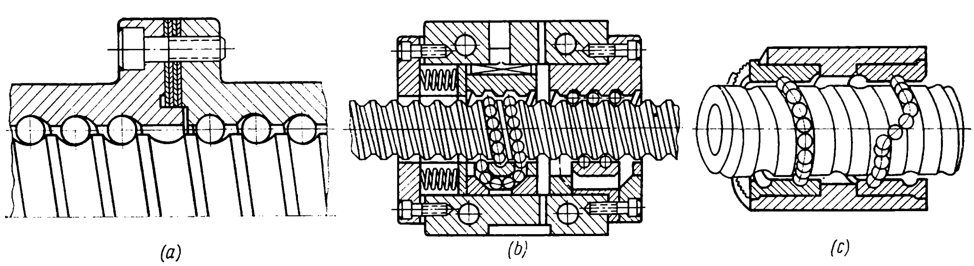


Fig. 87. Backlash elimination and preloading in ball-bearing screws and nuts:
(*a*) and (*b*) by axial displacement of the nut sections; (*c*) by relative rotation of the nut sections

2. *Rigidity calculations* based on the axial deflection of the nut relative to the screw due to elastic deformations in the threaded unit. Preloading substantially increases the contact rigidity and enables the deformation to be assumed as being proportional to the load.

The rigidity of ball-bearing screws and nuts, based on Std H23-7, is

$$j \cong 0.3z \sqrt[3]{d_1 P_{pr}} \text{ kgf per mm} \tag{146}$$

where d_1 = ball diameter, mm
z = design number of balls
P_{pr} = normal preload force per ball, kgf.

Taking errors in manufacture into consideration, the design number of balls can be taken equal to

$$z = 0.7 z_{nom} \tag{147}$$

where z_{nom} is the nominal number of balls carrying the load.

In addition to other factors, ball-bearing screws that are in continuous operation should be checked for durability.

6-5. Devices for Small Displacements

The rigidity of ordinary mechanisms of the rack and pinion or the screw type is often insufficient to provide very accurate small displacements.

Under definite conditions, a unit in slow traverse is subject to what has been called "stick-slip" phenomena which is a nonuniform motion with alternating stops and jumps. Stick-slip motion begins at speeds below the critical value for the given system of drive. Thus

$$v_{cr} = \frac{\Delta f N}{\sqrt{\psi k m}} \text{ m per sec} \tag{148}$$

where Δf = difference between the coefficients of sliding and static friction
N = normal force exerted on the ways, kgf
ψ = relative energy dissipation in vibrations (dimensionless value)
k = reduced rigidity of the drive, kgf per m
m = mass of the traversed unit, kg.

Under conditions of boundary friction, typical of machine tool slideways, $\Delta f = 0.05$ to 0.15, and the relative energy dissipation in vibrations is approximately $\psi = 1$ to 2.

The linear value of the jumps in this type of nonuniform motion is determined by the relationship

$$s = \lambda \frac{\Delta f N}{k} \text{ m} \tag{149}$$

where $\lambda \geqslant 1$ is a factor depending upon damping in the drive.

Nonuniform "stick-slip" motion can be avoided, or its detrimental effects can be reduced, either by improving the friction conditions (by using hydrostatic or aerostatic slideways), or by increasing the rigidity of the drive. This has led, in machine tool engineering, to the use of special devices operating without clearances or backlash and ensuring a high rigidity in the drive.

Thermal-Expansion Drives

The motive device in a thermal-expansion drive is the thermal element whose temperature deformation produces small displacements of the travelling unit without the need for any other kinematic links. The principle of this drive was developed by B. Breyev and has been applied in the infeed drive of a number of models of cylindrical grinders.

A thermal-expansion drive (Fig. 88) consists of a rigid hollow rod whose one end is secured to a stationary part of the machine (bed or base) while the other end is linked to the travelling unit. When the rod is heated, its free end has a displacement equal to

$$\Delta l_t = \alpha l \Delta t$$

where α = coefficient of linear expansion of the rod material
l = length of the rod at the initial temperature
Δt = temperature increment.

The rod is heated either by means of an electrical heater coil or by passing low-voltage high-amperage current through the rod itself (Fig. 89). The rod is cooled to return the unit to its initial position by passing cutting fluid from the coolant system through it.

One drawback of a thermal-expansion drive is that it evolves heat which may lead to temperature deformation of adjacent units, thereby reducing machining accuracy. Another drawback is due to thermal inertia which does not allow this drive to be used for frequently repeated movements.

Magnetostriction Drives

The operating principle of the magnetostriction drive (Fig. 90) is similar to that of the thermal-expansion drive. The required displacement is obtained by creating a magnetic field around the free end of a rod made of ferromagnetic material. The length of the rod is changed by varying the strength of the field (Fig. 91). Some materials expand when the field strength is increased (positive magnetostriction) while others contract (negative magnetostriction).

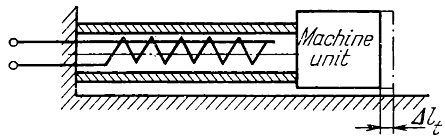


Fig. 88. Principle of the thermal-expansion drive

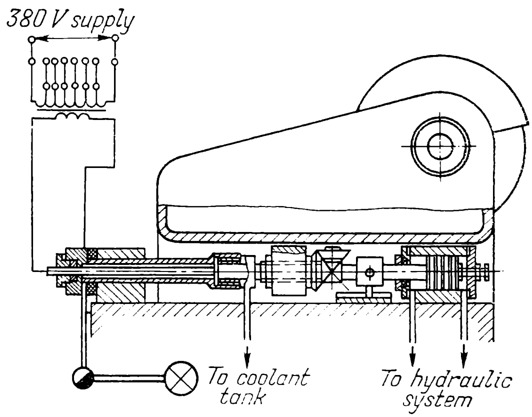


Fig. 89. A thermal-expansion drive in a grinder for wheelhead infeed

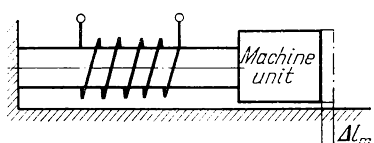


Fig. 90. Principle of the magnetostriction drive

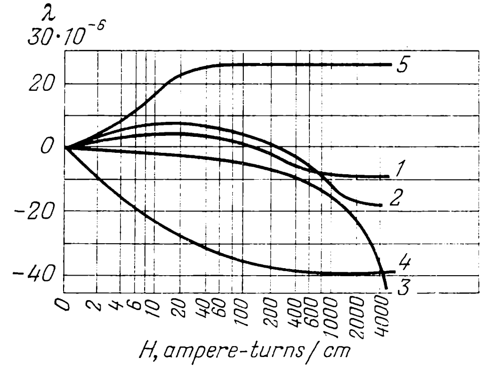


Fig. 91. Relative magnetostrictional expansion of rods of various materials:
1—iron; *2* and *3*—cobalt; *4*—nickel; *5*—permalloy

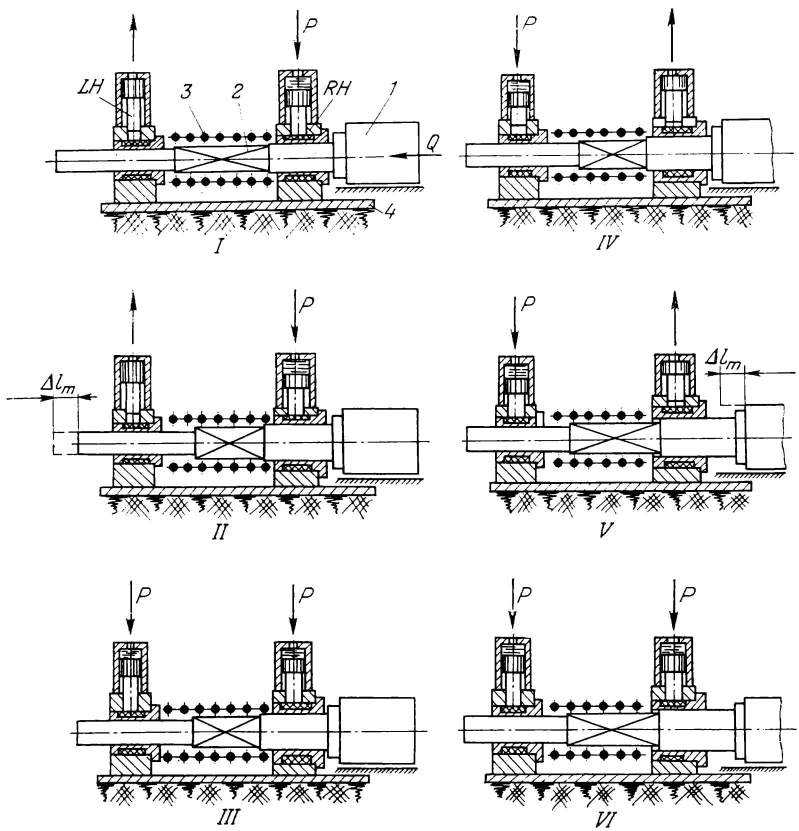


Fig. 92. Magnetostriction drive combined with an "inchworm" unit:
1—unit to be traversed; *2*—rod; *3*—coil; *4*—stationary part of the machine; *LH* and *RH*—left- and right-hand hydraulic clamps, respectively

The expansion of the rod under the action of magnetostriction is

$$\Delta l_m = \lambda l$$

where λ = relative magnetostrictional expansion
l = length of the rod.

This expansion of the rod and, consequently, the displacement of the machine unit, is limited for materials that can be feasibly used in practice by a very small value, in the order of 6 or 7 microns per 100 mm of rod length. This substantially restricts the possible applications of a magnetostriction drive.

To eliminate the above-mentioned shortcoming, the so-called "inchworm" unit has been combined with the magnetostriction drive. The principle of this device is illustrated in Fig. 92. It consists of right- (*RH*) and left-hand (*LH*) hydraulic clamps, coil *3* and rod *2* linked to the travelling unit *1*. In position *I*, *RH* is actuated (tightened), *LH* is released and the coil is energized. The coil is then de-energized and rod *2* contracts by the amount Δl_m (position *II*). Next, *LH* is actuated (position *III*) and *RH* is released (position *IV*). In position *V*, the coil is energized, expanding the rod and moving the unit by the amount Δl_m. Finally, *RH* is actuated (position *VI*) and then *LH* is released (position *I* again). This sequence of clamping and unclamping combined with magnetostriction action can be repeated until the required length of travel is obtained. The application of an inchworm drive, however, complicates the construction of the machine tool unit and leads, inevitably, to a loss in rigidity.

Elastic-Link Drives

The deformation of a component (the elastic link) connected to a travelling unit can be used to obtain small movements of a magnitude comparable with elastic displacements. An elastic link in the form of a flat or leaf spring can be employed for relatively large linear displacements. A drive with an elastic link for wheelhead infeed in a grinder is shown schematically in Fig. 93. Here, the leaf spring is first bent by means of the hydraulic system. Then, as the oil drains freely from the cylinder through an aperture of small cross section, the spring straightens out and its free end moves the wheelhead.

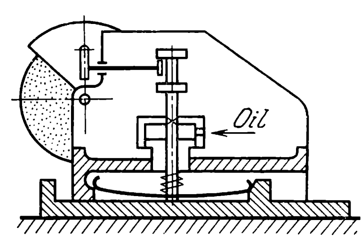


Fig. 93. Principle of the elastic-link drive

Drawbacks of elastic-link drives are the restricted amount of displacement (within the limits of elastic deformations) and, as a rule, the variable rigidity.

CHAPTER 7

MECHANISMS FOR PERIODIC (INTERMITTENT) MOTION

7-1. Periodic Motions in Machine Tools and Devices for Producing Them

In certain machine tools the working process is devised so that the relative positions of the blank and cutting tools must be changed periodically in order to obtain the finished workpiece. Such a periodic movement of a unit or part of the machine tool may occur before a new stroke, pass or cycle and may be either rectilinear and of definite length each time, or rotary and through a definite angle. Periodic motions of the latter type are called *indexing motions*.

Periodic motions include, for instance, the feed motions in planers, shapers and slotters, infeed in grinders, turret indexing and various movements in automatic and semiautomatic machine tools operating on a cycle.

Requirements made to the accuracy of periodic movements depend upon the specified dimensional accuracy and surface finish of the work that is to be machined. In this respect, highest accuracy is required of the mechanisms for indexing spindle carriers, multiple-station tables and lathe turrets, and of the indexing devices of gear-cutting machines operating by the single-indexing principle. On the other hand, no especial accuracy of motion is required of the feed mechanisms of planers, shapers and slotters.

Regardless of its construction, a device effecting a displacement of some unit of a machine tool cannot in itself guarantee high accuracy of periodic motions nor their repeatability. These factors are determined by the errors in manufacturing and assembling the mechanism, clearances in its mating components, actions of inertia forces, etc. Hence, to obtain highly accurate periodic motions, it is necessary to make provision for automatic *locking mechanisms* that can ensure accurate fixed positioning of the unit being displaced at the end of each movement.

Periodic motions are effected, in modern machine tools, by: (1) various types of cam mechanisms, (2) mechanisms incorporating overrunning clutches, (3) ratchet gearing mechanisms, (4) Geneva wheel mechanisms, and (5) electric, hydraulic and pneumatic mechanisms. Magnetostriction devices (see p. 152) and step motors have also been used to some extent for this purpose. The latter are especially promising since they enable the periodic feed to be varied in quite a wide range, and also to change it automatically in accordance with the cutting speed.

One example of cams used for periodic motions is the single-, two- and three-pass plate cams of gear shapers. It is frequently difficult to employ cams to obtain motions of considerable overall length.

An overrunning clutch can be conveniently applied in periodic motion trains where the first driving link of the train has a reciprocating motion. When this link moves in one direction the overrunning clutch provides a rigid and positive kinematic linkage between the corresponding elements of the train; upon movement of the link in the opposite direction, the clutch is disengaged and the linkage is eliminated.

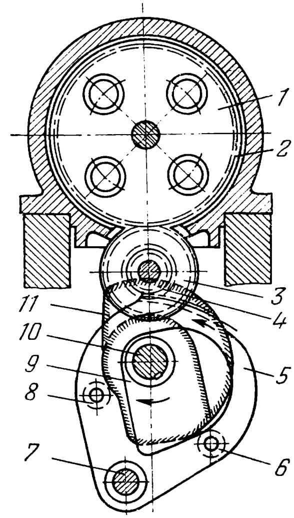


Fig. 94. Spindle carrier indexing mechanism of a four-spindle automatic bar machine

Both cams and an overrunning clutch are employed in a mechanism for indexing the carrier of a four-spindle automatic bar machine (Fig. 94). Cams *9* and *11* are keyed to the camshaft *10* of the automatic and are in constant contact with rollers *6* and *8* of the segment gear *5*. The cams are profiled in such manner that segment gear *5* accomplishes a periodic rocking motion on pin *7*. This motion is transmitted through gears *3* and *4* to gear rim *2* of carrier *1*. An overrunning clutch is built into the common hub of gears *3* and *4*. As a result, carrier *1* is turned (indexed) in only one direction—counterclockwise.

7-2. Ratchet Gearing Mechanisms

Ratchet gearing is especially convenient in cases when the time allotted to the displacement is limited. For this reason, it is frequently applied in the feed mechanisms of machine tools in which the intermittent feed movement takes place during the overtravel of the tool or the rapid return stroke (in planers, shapers, slotters, grinders and gear finishing machines).

In most cases ratchet gearing is used to obtain rectilinear motion of the corresponding unit. In this case, a pawl periodically turns a ratchet wheel with external or internal teeth through a definite angle. The ratchet wheel is linked kinematically to a power screw which traverses the table, slide, etc. Rotary periodic motions can also be accomplished by means of ratchet gearing.

In one full stroke (back and forth) of the pawl, the ratchet wheel can be turned through an angle as large as 90° or 100°, but in most cases the angle does not exceed 45°.

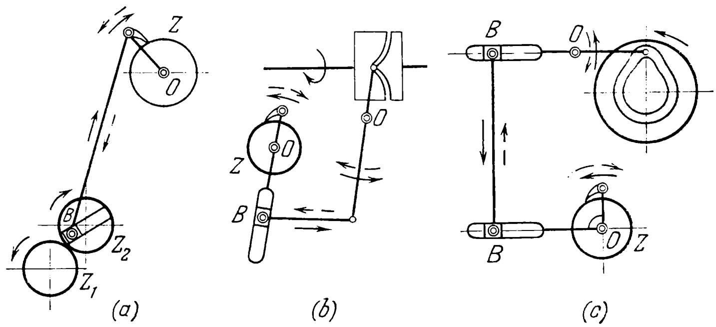


Fig. 95. Varying periodic (intermittent) motions produced by ratchet gearing

The amount of intermittent motion produced by ratchet gearing should, as a rule, be variable. The motion can be varied: (a) by changing the angle of swinging movement of the arm that carries the operating pawl or (b) if the arm oscillates through a constant arc, by covering the ratchet wheel teeth over a part of the arc described by the pawl, or by automatically lifting the pawl out of engagement during part of its stroke.

Mechanical versions of the first of these principles are shown schematically in Fig. 95*a*, *b* and *c*. The angle of oscillation of the pawl arm is varied by adjusting slide block *B* along the slot of a crank disk (Fig. 95*a*) or of rocker arms (Fig. 95*b* and *c*). In hydraulically operated machine tools, the swing of the pawl is varied by changing the stroke of the piston actuating the pawl.

The principle of devices that vary the angle of ratchet wheel rotation when the pawl stroke remains constant can be seen in Fig. 96. Here shield *1* can be

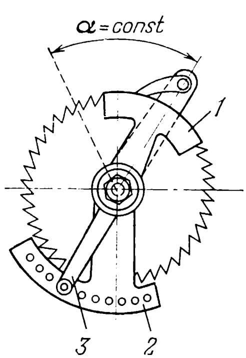


Fig. 96. Variable-motion ratchet gearing

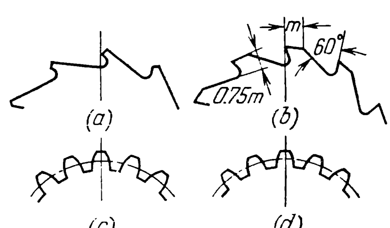


Fig. 97. Tooth forms of ratchet wheels employed in machine tools

adjusted to cover more or less teeth of the ratchet wheel within angle α. The shield is held in the required position by the spring-loaded plunger of lever *3* that engages a hole of stationary sector *2*.

If the intermittent motion of the ratchet wheel is to be reversible (as, for instance, in the feed mechanisms of planers, shapers and slotters) the teeth must be of symmetrical form and the pawl must be designed so that it properly engages the teeth after being turned over to operate in the reverse direction.

The number of teeth z of the ratchet wheel is determined from kinematic calculations of the train; in the great majority of cases $z = 12$ to 250. The circular pitch of the teeth is $t = \pi m$, where m, the module, is selected so that the diameter of the wheel is not too large for the unit of which it is to be a component. Standard H22-4, worked out by ENIMS, stipulates the following ranges for external ratchet gearing: $z = 20$ to 200, $m = 0.6$ to 2.5 mm and wheel pitch diameter $D = 30$ ($zm = 50 \times 0.6$) to 200 ($zm = 200 \times 1$) mm and for internal ratchet gearing: $z = 24$ to 200, $m = 0.6$ to 2.5 mm and $D = 60$ to 200 mm.

Tooth forms of ratchet wheels are shown in Fig. 97 in which *a* and *b* are for nonreversible gearing and *c* and *d* for reversible gearing. The working flank of the teeth of nonreversible ratchet wheels should be either radial or slightly undercut (10° in Standard H22-4).

The possibilities offered by the application of ratchet gearing to produce periodic motions in machine tools are illustrated by Fig. 98 which shows the indexing device of the model 345 semiautomatic spline grinder. Here the swing of the pawl remains constant (100°) while the periodic rotation of the ratchet wheel, linked to the work spindle, is varied by covering teeth with a shield.

This indexing device operates as follows. Before the working stroke of the table (to the right), oil under pressure is admitted into the right end of cylinder *10* forcing plunger *11* to the left. Rack teeth, cut on plunger *11*, mesh with segment gear *9* which carries pawl *7*. The sector begins to turn clockwise with the pawl sliding over the periphery of shield *8*. The cam lobe of segment gear *9* actuates roller *3*, withdrawing locking plunger *4* from a slot of index plate *1* which is secured to ratchet wheel *6*. This releases the ratchet wheel. Then pawl *7* runs off shield *8* and engages a tooth of the ratchet wheel, turning it through the required angle. The index plate, work spindle and spline shaft being ground are turned together with the ratchet wheel. Somewhat before the end of this motion, the cam of segment gear *9* releases plunger *4* which, under the pressure of the oil in cylinder *5*, enters the next slot of the index plate.

The mechanism is returned to its initial position by switching the oil flow to the left end of cylinder *10*. Dog *2*, clamped on index plate *1*, operates a small-size limit switch which transmits a command to the infeed mechanism

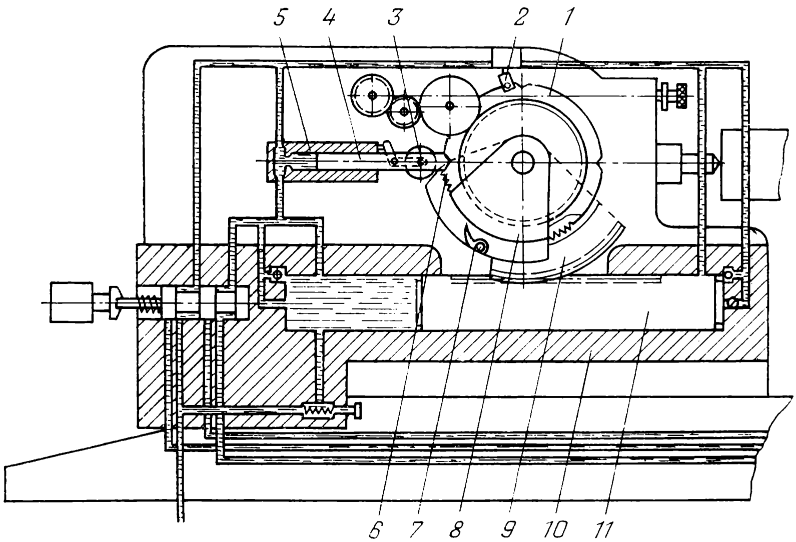


Fig. 98. Indexing device of the model 345 semiautomatic spline grinder

of the grinding wheel after each pass over all the teeth (splines), i.e., after each complete revolution of the work spindle together with the index plate.

Ratchet gearing also finds application in counting mechanisms of machine tools, including devices for automatically switching off the machine after a certain definite element has completed a preset number of full strokes or revolutions.

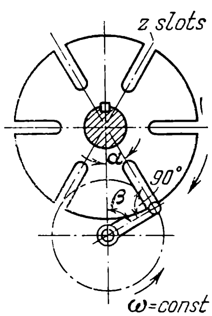


Fig. 99. Diagram of a flat Geneva wheel mechanism with external engagement

7-3. Geneva Wheel Mechanisms

A Geneva wheel mechanism, consisting of a driver and the wheel proper (Fig. 99), differs from ratchet gearing in that the angle of periodic rotation cannot be varied. Consequently, Geneva mechanisms are used in machine tools mainly in indexing devices with a constant angle of periodic rotation. These applications include the indexing of spindle carriers in automatic and semiautomatic lathes, of turrets and of multiple-station rotary tables, etc.

If a transmission with a variable ratio (for example, change gears) is introduced in the kinematic train

between the Geneva wheel and the indexed component of the machine tool, the angle of rotation of this component can be varied though the Geneva wheel rotates periodically through a constant angle.

With rare exceptions, only normal flat Geneva wheel mechanisms with equal angles between the adjacent radial slots (Fig. 99) and with external engagement are found in modern machine tools. Hence, only this type will be considered in the following.

Principal Kinematic Relationships

Let us assume that the driver of the Geneva wheel, which is usually designed as a lever or crank with a roller or, less frequently, a pin at the end or even as a pin wheel, rotates at a constant angular velocity $\omega = \frac{\pi n}{30}$ sec^{-1}, where n is the speed, rpm, of the driver shaft. The rotation of the Geneva wheel through the angle 2α (between adjacent slots) takes place during the time the driver rotates through the angle 2β. During the rest of the revolution of the driver, through the angle $2(\pi - \beta)$, the Geneva wheel is at rest.

If T is the time required for a full revolution of the driver, t_i is the time required for indexing the Geneva wheel and $t_r = (T - t_i)$ is the time the wheel is at rest, then, for $\omega =$ const we can write

$$\frac{t_i}{T} = \frac{2\beta}{2\pi} = \frac{\beta}{\pi} \quad \text{and} \quad \frac{t_r}{T} = \frac{2(\pi - \beta)}{2\pi} = 1 - \frac{\beta}{\pi} \tag{150}$$

To avoid impacts at the beginning and end of the Geneva wheel motion, the mechanism should be designed so that the angle between the driver and the slot is 90° when the roller enters and leaves the slot. In this case $\alpha + \beta = \frac{\pi}{2}$ and $\beta = \frac{\pi}{2} - \alpha = \frac{\pi(z-2)}{2z}$, where $\alpha = \frac{\pi}{z}$, z being the number of slots in the Geneva wheel.

Hence

$$\frac{t_i}{T} = \frac{\beta}{\pi} = \frac{z-2}{2z} \quad \text{and} \quad \frac{t_r}{T} = 1 - \frac{t_i}{T} = \frac{z+2}{2z} \tag{151}$$

or, since $T = \frac{60}{n}$ sec,

$$\left.\begin{aligned} t_i &= \frac{z-2}{2z} T = \frac{z-2}{z} \frac{30}{n} \text{ sec} \\ t_r &= \frac{z+2}{z} \frac{30}{n} \text{ sec} \end{aligned}\right\} \tag{152}$$

and the working time coefficient of the Geneva wheel is

$$k = \frac{t_i}{t_r} = \frac{z-2}{z+2} \tag{153}$$

If the time t_r that the wheel is to be at rest is given, the required speed of the driver shaft will be

$$n = \frac{z+2}{z} \frac{30}{t_r} \text{ rpm} \tag{154}$$

The loss in productivity due to the periodic indexing of the Geneva wheel can be reduced by reducing the time t_i. Since the time t_r depends upon the processing operation, t_i can be reduced if $\omega = \text{const}$ only by reducing the number of slots in the Geneva wheel and adding transmissions in the kinematic train to obtain the required number of stations of the component being indexed. Such a solution is undesirable and, in some cases, unacceptable since, other things being equal, a reduction in the number of slots leads to an increase in the inertia torques acting on the driver and wheel.

Another method of decreasing the time t_i is to increase the speed of the driver shaft during the period of Geneva wheel rotation. This possibility, however, is limited by the increase it leads to in inertia torques.

A more favourable relation between t_i and t_r can be obtained by stopping the driver or slowing it down during the time that the part being indexed is to be at rest, and automatically engaging driver rotation just before indexing is to take place. In this case, the angular velocity of the driver can be assigned so high that the time t_i will be sufficiently short. Such a solution is applied in unit-built machine tools in which the indexing of a multiple-station table is powered from a separate electric motor through a Geneva wheel mechanism.

If the kinematic constraint between the camshaft and the driver is not to be disengaged, the problem can be solved by including a transmission between them which must satisfy two conditions: (a) the driver shaft must make one full revolution for each full revolution of the camshaft; and (b) during the time that the camshaft turns through a certain angle δ, reserved for indexing, the driver shaft turns through the angle 2β.

In principle, this can be effected by any transmission having an average transmission ratio $i_{av} = 1$ and whose driven element rotates at variable speed when the driving element has constant angular velocity, for instance, elliptical gearing.

Another solution is to combine the Geneva wheel mechanism with intermittent gearing, link or other mechanism. These devices are difficult to manufacture and unreliable in operation and therefore have been used very seldom in machine tool design (see p. 173).

Since the time required to turn the Geneva wheel $t_i > 0$, it follows from equation (151) or (152) that the wheel can not have less than three slots.

We can write for the random position of the Geneva wheel mechanism shown in Fig. 100 that

$$\tan \psi = \frac{\lambda \sin \varphi}{1 - \lambda \cos \varphi} \tag{155}$$

where $\lambda = \frac{r}{e}$

r = driver radius

e = centre-to-centre distance.

Hence the velocity of the Geneva wheel is

$$\omega_w = \frac{d\psi}{dt} = \frac{\lambda(\cos\varphi - \lambda)}{1 - 2\lambda\cos\varphi + \lambda^2}\,\omega \tag{156}$$

where ω is the angular velocity of the driver or pin wheel and is constant during Geneva wheel rotation.

The angular acceleration of the Geneva wheel is

$$\varepsilon_w = \pm\frac{\lambda(1-\lambda^2)\sin\varphi}{(1-2\lambda\cos\varphi+\lambda^2)^2}\,\omega^2 \tag{157}$$

in which $\varepsilon_w > 0$ for the first half of the wheel motion where its angular velocity increases, and $\varepsilon_w < 0$ for the second half (Fig. 101).

To avoid a solid impact at the beginning of wheel rotation, when the pin engages the slot (the position of the mechanism shown with chain lines in Fig. 100), it is necessary that $\omega_{w\ in} = 0$, i.e., as follows from equation (156), for $\varphi = \beta$, the condition that $\cos\beta - \lambda = 0$ must be satisfied. Hence, $r = e\cos\beta$ or

$$\lambda = \frac{r}{e} = \sin\alpha = \sin\frac{\pi}{z} \tag{158}$$

which means that the pin (or roller) must enter the slot in the radial direction (see Fig. 99).

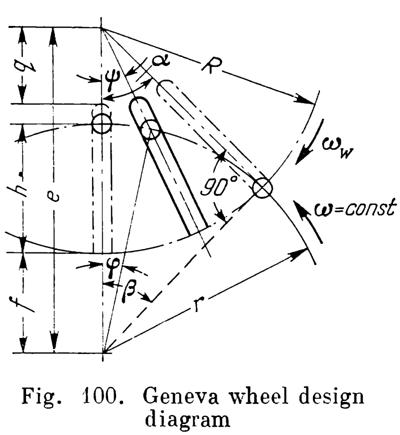


Fig. 100. Geneva wheel design diagram

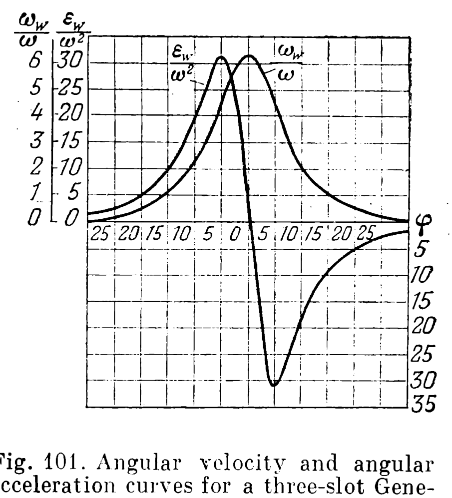


Fig. 101. Angular velocity and angular acceleration curves for a three-slot Geneva wheel with external engagement

In this case, equations (156) and (157) become

$$\omega_w = \frac{\sin\frac{\pi}{z}\left(\cos\varphi - \sin\frac{\pi}{z}\right)}{1-2\sin\frac{\pi}{z}\cos\varphi+\sin^2\frac{\pi}{z}}\,\omega \tag{159}$$

$$\varepsilon_w = \pm\frac{\sin\frac{\pi}{z}\cos^2\frac{\pi}{z}\sin\varphi}{\left(1-2\sin\frac{\pi}{z}\cos\varphi+\sin^2\frac{\pi}{z}\right)^2}\,\omega^2 \tag{160}$$

Plotted on the basis of these equations, the curves of ω_w and ε_w (or of the ratios $\frac{\omega_w}{\omega}$ and $\frac{\varepsilon_w}{\omega^2}$ which are numerically equal to ω_w and ε_w, respectively) are for a three-slot Geneva wheel with a driver velocity $\omega = 1\ \text{sec}^{-1}$.

The above formulas for the angular velocity ω_w and angular acceleration ε_w of a Geneva wheel refer to a theoretical case in which $\omega = \text{const}$ during wheel indexing, the relationship (158) is accurately maintained, the centre lines of the Geneva wheel slots are strictly radial, the angles between adjacent slots are exactly equal, the pin or roller of the driver is fitted to the slots without clearance, etc. Deviations from such an ideal case are inevitable in practice. As a result, curves ω_w and ε_w, when obtained experimentally, differ to a more or less considerable extent from the theoretical curves.

The lack of constancy of ω_w during wheel indexing and the inertia effects this leads to are the cause of premature wear of the parts of the indexed unit. This is one of the drawbacks of a Geneva wheel mechanism.

As is evident from equations (156) and (157), $\omega_w = \max$ at $\varphi = 0$ at the middle point in wheel indexing. Substituting this value of φ in equation (156) we can write

$$\omega_{w\,max} = \frac{\lambda(1-\lambda)}{1-2\lambda+\lambda^2}\,\omega = \frac{\lambda}{1-\lambda}\,\omega = \frac{\sin\frac{\pi}{z}}{1-\sin\frac{\pi}{z}}\,\omega \tag{161}$$

Therefore, the fewer the slots in the Geneva wheel, the higher the maximum angular velocity $\omega_{w\,max}$ of the wheel at the same angular velocity ω of the driver.

The angular acceleration of the Geneva wheel at the beginning and end of the indexing motion are determined from equation (160) for $\varphi = \beta = \frac{\pi}{2} - \alpha = \frac{\pi}{2} - \frac{\pi}{z}$ as

$$\varepsilon_{w\,in,\,fin} = \pm\frac{\sin\frac{\pi}{z}\cos^3\frac{\pi}{z}}{\left(1-2\sin^2\frac{\pi}{z}+\sin^2\frac{\pi}{z}\right)^2}\,\omega^2 = \pm\,\omega^2\tan\frac{\pi}{z} \tag{162}$$

Since, in all cases, $\tan\frac{\pi}{z} > 0$, then $\varepsilon_{w\ in,\ fin} \neq 0$. This means that Geneva wheel indexing is always accompanied by impact load increase at the initial moment ($\omega_w = 0$, $\varepsilon_w \neq 0$). The fewer the number of slots, the heavier the impact will be, as is evident from the last equation. Therefore, from the point of view of prolonging the service life of the locking devices, it proves more advantageous to apply Geneva wheel mechanisms with a large number of slots.

At the middle point in indexing $\varphi = 0$, and it follows from equation (157) or (160) that in this position $\varepsilon_w = 0$ because the sign of the angular acceleration changes (see Fig. 101).

The maximum angular acceleration $\varepsilon_{w\ max}$ of a Geneva wheel increases rapidly with a reduction in the number of its slots. For example, the maximum acceleration of a three-slot wheel is approximately 23-fold that of a six-slot wheel for the same angular velocity of the drivers. Hence, Geneva wheel mechanisms with a small number of wheel slots are not advantageous in respect to their dynamic performance. For this reason, in designing a Geneva wheel mechanism for indexing a three- or four-station table, head or other unit, it frequently proves expedient to use a five-, six- or even eight-slot Geneva wheel and to arrange a transmission with the required ratio between the wheel and the unit to be indexed.

The requirement that the indexing of the Geneva wheel should not be accompanied with a solid impact is primary in determining the relations between the geometrical dimensions of the mechanism. In addition to the relationship expressed by equation (158), it follows from Fig. 100 that

$$\lambda_1 = \frac{R}{e} = \cos\frac{\pi}{z} = \sqrt{1-\lambda^2} \tag{163}$$

This indicates that only one of the dimensions, r, R or e, can be assigned at will.

The length of the slot should be somewhat larger than

$$h = r + R - e = e\left(\sin\frac{\pi}{z} + \cos\frac{\pi}{z} - 1\right) \tag{164}$$

In order to enable the driver or pin wheel to be secured on a shaft between bearings on both sides of the Geneva wheel, the diameter of this shaft must comply with the condition

$$d < 2f = 2(e - R) = 2e\left(1 - \cos\frac{\pi}{z}\right)$$

or, otherwise,

$$\frac{d}{e} < 2\left(1 - \cos\frac{\pi}{z}\right) = 4\sin^2\frac{\pi}{2z} \tag{165}$$

At large values of z, the ratio $\frac{d}{e}$ is small and, to avoid increasing the distance e excessively, it often becomes necessary to mount the driving element of the mechanism at the end of its shaft, i.e., overhanging the bearing.

A condition similar to (165) is

$$\frac{d_w}{e} < 2\left(1 - \sin\frac{\pi}{z}\right) = 4\sin^2\left(\frac{\pi}{4} - \frac{\pi}{2z}\right) \tag{166}$$

where d_w is the diameter of the shaft on which the Geneva wheel is secured.

Constructional Features

The construction of a Geneva wheel mechanism depends upon the accepted kinematic scheme and the permissible overall size.

The driving element can be designed in the form of a lever, pin wheel or a gear or worm wheel carrying the pin tooth, which can be a roller (bushing) mounted either directly on the pin or on needle rollers. In some cases a ball bearing of suitable diameter, mounted on the pin, serves as the pin tooth. Both single-support (overhanging) and two-support rollers are used as pin teeth in the Geneva wheel mechanisms of machine tools. The second, more rigid design is to be preferred.

The driven element is made either as a solid part in the form of a wheel or disk, or it is assembled of separate sectors or strips fastened to the part to be indexed in such manner that the spaces between them constitute the slots of the Geneva wheel.

The rollers are made (in the USSR) of steel grade ШХ15 hardened to 59-63R_C or of steel 20X which is carburized and then hardened to 56-62R_C.

In the Geneva wheel, the components subject to wear are usually made of steel 40X hardened to 45-50R_C.

Examples illustrating the applications of Geneva wheel mechanisms can be found in Sec. 2-8, Vol. 4.

Design of Flat Geneva Wheel Mechanisms with External Engagement

Calculations involved in the design of a Geneva wheel mechanism include the determination of the power required to index the wheel, of contact stresses of the components of the mechanism and the bending stress of the driver pin (or roller axle).

Precise calculations are complicated by the variable efficiency η of a Geneva wheel mechanism. However, the assumption that $\eta = \text{const}$ gives results that are sufficiently accurate for all practical purposes.

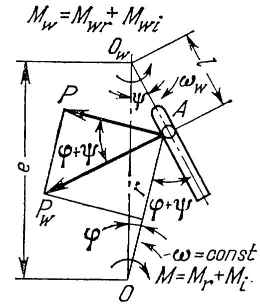


Fig. 102. Geneva wheel design diagram

In each position of the wheel during indexing (Fig. 102), its shaft is subject to the torque M_{wr}, due to the resistance to motion of the masses linked to the wheel, and the torque M_{wi} of the inertia forces, resulting from the fact that $\omega_w \neq$ const. The total torque acting on the wheel shaft is

$$M_w = M_{wr} + M_{wi} = M_{wr} + I\varepsilon_w \tag{167}$$

where I = moment of inertia of the displaced masses referred to the shaft of the Geneva wheel

ε_w = angular acceleration of the Geneva wheel.

It can be assumed that M_{wr} = const.

Consequently, the torque on the shaft of the driver in indexing the Geneva wheel is

$$M = M_w \frac{\omega_w}{\omega}\frac{1}{\eta} = (M_{wr} + I\varepsilon_w)\frac{\omega_w}{\omega}\frac{1}{\eta} \tag{168}$$

This torque can be thought of as the algebraic sum of the torques

$$\left.\begin{aligned} M_r &= M_{wr}\frac{\omega_w}{\omega}\frac{1}{\eta} \\ \text{and} \qquad & \\ M_i &= M_{wi}\frac{\omega_w}{\omega}\frac{1}{\eta} = I\,\frac{\varepsilon_w\omega_w}{\omega}\frac{1}{\eta} \end{aligned}\right\} \tag{169}$$

of which the first is a result of the moment of the static forces of resistance to motion, referred to the wheel shaft, while the second is the moment of the inertia forces. Usually $M_r \ll M_i$.

After selecting a value of η (see below), equations (156) and (157) can be used to determine the values of ω_w and ε_w for angles φ of driver rotation from $\beta = \frac{\pi}{2} - \frac{\pi}{z}$ to 0, so as to plot the curves for M_{wi}, M_r and M_i which will provide a sufficiently complete idea of these torques for various positions of the mechanisms if $\Delta\varphi$ is taken in intervals of about 3° to 5°. Since it is assumed that M_{wr} = const and that ω = const as well, the ordinates of the curves for M_{wi} and M_r are proportional, respectively, to the ordinates of the curves for ε_w and ω_w (see Fig. 101). The expression for the torque M_i includes the product $\varepsilon_w\omega_w$ and therefore it varies according to a more complicated law than does M_r and, in contrast to the latter, has different signs in the first and second halves of the indexing motion.

The instantaneous power on the driver shaft is

$$\left.\begin{aligned} N &= M\omega \text{ kgf-mm per sec} \\ \text{or} \qquad N &= \frac{M\omega}{102,000} \text{ kW} \end{aligned}\right\} \tag{170}$$

As can be seen from equation (169) for M_i, the average torque $\overline{M}_i = 0$. Consequently, the total average torque $\overline{M}$ acting on the driver shaft, is equal to the average value $\overline{M}_r$ of the torque due to the static forces of resistance. Thus

$$\overline{M} = \overline{M}_r = \frac{1}{2\beta} 2 \int_0^\beta M_r \, d\varphi \tag{171}$$

Here, we substitute for M_r its value from equation (169) and, since $M_{wr} = \text{const}$ and $\frac{\omega_w}{\omega} = \frac{d\psi}{d\varphi}$, we obtain

$$\overline{M} = \frac{1}{\beta} \frac{M_{wr}}{\eta} \int_0^\beta \frac{d\psi}{d\varphi} \, d\varphi = \frac{1}{\beta} \frac{M_{wr}}{\eta} \int_0^\alpha d\psi$$

(see Fig. 102). Now, since $\alpha = \frac{\pi}{z}$ and $\beta = \frac{\pi(z-2)}{2z}$ (see page 161), the final expression will be

$$\overline{M} = \frac{2}{z-2} M_{wr} \frac{1}{\eta} \tag{172}$$

The average power required by the driver should be calculated, however, for the first half of the indexing motion when the load is higher because the masses being indexed are being accelerated ($\varepsilon_w > 0$). During this part of the indexing motion the average torque is

$$\overline{M}' = \frac{2}{z-2} \left[M_{wr} + \frac{zI}{2\pi} \left(\frac{\lambda}{1-\lambda} \right)^2 \omega^2 \right] \frac{1}{\eta} \tag{173}$$

Therefore, the average torque $\overline{M}'_i$, resulting from the inertia forces, depends, apart from the referred moment of inertia I of the displaced masses and the angular velocity ω of the driver, only upon the number of slots in the Geneva wheel $\left(\lambda = \sin \frac{\pi}{z} \right)$. Thus

$$\overline{M}'_i = \frac{z}{z-2} \frac{I}{\pi} \left(\frac{\lambda}{1-\lambda} \right)^2 \omega^2 \frac{1}{\eta} = \frac{z}{z-2} \frac{\pi}{900} \left(\frac{\lambda}{1-\lambda} \right)^2 In^2 \frac{1}{\eta} \tag{174}$$

After determining $\overline{M}'$ by means of equation (173), we can find the average power required to drive the indexing device in the first half of the

indexing motion. Thus

$$\overline{N}=\overline{M}'\omega \text{ kgf-mm per sec or } \overline{N}=\frac{\overline{M}'\omega}{102{,}000} \text{ kW} \tag{175}$$

If the Geneva wheel mechanism is powered from a separate electric motor (as, for instance, in machine tools with multiple-station tables) it is necessary in selecting the motor to take into account, in addition to the average power $\overline{N}$, the maximum power N_{max} required during Geneva wheel indexing, on the one hand, and the capability of the motor to withstand short overloads, on the other hand. The determination of the maximum torques $M_{w\ max}$ and M_{max} acting on the wheel and driver shafts, respectively, is also necessary in designing the parts of the mechanism.

Evidently

$$M_{w\ max}=M_{wr}+M_{wi\ max}$$

in which case $M_{wr}=\text{const}$ and

$$M_{wi\ max}=I\varepsilon_{w\ max}=I\frac{\varepsilon_{w\ max}}{\omega^2}\left(\frac{\pi n}{30}\right)^2=0.011\frac{\varepsilon_{w\ max}}{\omega^2}In^2 \tag{176}$$

The maximum component torques acting on the driver shaft are

$$\left.\begin{aligned} M_{r\ max}&=M_{wr}\frac{\omega_{w\ max}}{\omega}\frac{1}{\eta}\\ M_{i\ max}&=\frac{I}{\omega}(\varepsilon_w\omega_w)_{max}\frac{1}{\eta}\end{aligned}\right\} \tag{177}$$

From this it follows that torque M_r reaches its maximum value in the middle position of the Geneva wheel during the indexing motion ($\varphi=0$), when $\omega_w=\omega_{w\ max}$. The torque $M_{r\ max}$ can be readily calculated using the relation $\frac{\omega_{w\ max}}{\omega}$.

The torque M_i, resulting from the inertia forces, reaches its maximum value at an angle φ which satisfies the conditions

$$\left.\begin{aligned} &\frac{d}{d\varphi}(\varepsilon_w\omega_w)=0\\ \text{and}\quad &\frac{d^2}{d\varphi^2}(\varepsilon_w\omega_w)<0\end{aligned}\right\} \tag{178}$$

After determining the value of the angle $\varphi=\varphi_m$, at which M_i is at its maximum, from the first of the preceding equations, the second can be readily calculated. Thus

$$M_{i\ max}=\frac{\lambda^2(1-\lambda^2)(\cos\varphi_m-\lambda)\sin\varphi_m}{(1-2\lambda\cos\varphi_m+\lambda^2)^3}I\omega^2\frac{1}{\eta} \tag{179}$$

The torques M_r and M_i reach their maximum values at different values of angle φ. If the curves of these torques have been plotted, then $M_{max} = (M_r + M_i)_{max}$ is determined by plotting the resulting curve $(M_r + M_i)$ as a function of angle φ for the first half of the indexing motion. If these curves have not been plotted, M_{max} can be determined as the sum of $M_{i\ max}$ and the value of M_r corresponding to angle φ_m, since $M_{r\ max} \ll M_{i\ max}$ in all cases. The ensuing error is insignificant.

The maximum power required on the shaft of the driver is

$$\left.\begin{aligned} N_{max} &= M_{max}\omega \text{ kgf-mm per sec} \\ \text{or} \quad N_{max} &= \frac{M_{max}\omega}{102{,}000} \text{ kW} \end{aligned}\right\} \tag{180}$$

The ratio $\frac{N_{max}}{N} \cong \frac{M_{i\ max}}{\overline{M}_i'}$ is the larger, the fewer the number of slots. It reaches approximately 4 at $z = 3$. Therefore, the power peak in the drive of the Geneva wheel mechanism cannot be disregarded. The overload capacity (stalling torque ratio) of standard three-phase induction motors used in the Soviet machine tool industry is

$$U = \frac{M_{max}}{M_{nom}} \geqslant 1.8 \text{ to } 2$$

Thus, for example, a motor can be selected for a mechanism with a six-slot Geneva wheel having a nominal power rating only slightly larger (about 15 to 20 per cent) than that which corresponds to the average power required to index the wheel and the parts of the machine tool linked to the wheel. A motor with such a nominal power rating can safely withstand a peak load which is twice the average load. On the other hand, the nominal power rating of a motor for a three-slot Geneva wheel mechanism should be at least 2.2- or 2.3-fold that which corresponds to the calculated average power.

The actual peak load of the electric motor is somewhat less than the calculated value due to the inertia forces of the components of the drive up to the driver of the Geneva wheel.

Forces P and P_w (see Fig. 102), required in designing the components of the Geneva wheel mechanism, are determined from the corresponding torques. Thus

$$P_w = \frac{M_w}{l} \text{ and } P = \frac{M}{r} \tag{181}$$

in which

$$l = \sqrt{e^2 - 2er\cos\varphi + r^2} = \frac{r}{\lambda}\sqrt{1 - 2\lambda\cos\varphi + \lambda^2}$$

From this the values of the forces P and P_w can be determined in the form of the functions $P = F_1(M_w, \varphi)$ and $P_w = P_{wr} + P_{wi} = F_2(M_w, \varphi)$. Then the curves P_{wr} and P_{wi} can be plotted according to the angle φ.

To avoid complications in designing the components of a Geneva wheel mechanism and to take the variable nature of forces P and P_w into consideration, calculations are usually based on the maximum values of these forces. Force P_{wr}, due to the torque M_{wr} of the static forces of resistance to wheel indexing, reaches its maximum value at the middle of the indexing motion when the arm of moment M_{wr} is shortest (see Fig. 102). Thus

$$l_{min} = e - r = (\lambda^{-1} - 1)\, r$$

In this position of the mechanism

$$P_{wr\,max} = \frac{M_{wr}}{l_{min}} = \frac{M_{wr}}{r}\,\frac{\lambda}{1-\lambda} \tag{182}$$

Force P_{wi}, due to the inertia torque, reaches its maximum value at values of angle φ that depend upon the number of wheel slots z. For various values of z:

z	3	4	5	6	8
$P_{wi\ max}\colon \frac{In^2}{r}$ . .	1.966	0.126	0.0318	0.0131	0.00424

In this way, forces P_{wr} and P_{wi} reach their maximum values in different positions of the mechanism. Calculations for determining the maximum value of P_w can be simplified and results sufficiently accurate for practical purposes can be obtained if it is assumed that $P_w = P_{w\,max}$ when $P_{wi} = P_{wi\,max}$.

The calculated values of P_w and P are used to check the bearing strength of the roller and the working surfaces of the slot, and to design the bearings of the wheel and driver shafts.

The following average values (over one indexing motion) can be taken as the efficiency of Geneva wheel mechanisms: if the wheel is mounted on a shaft running in sleeve bearings—$\eta \cong 0.8$ to 0.9; if the shaft runs in anti-friction bearings—$\eta \cong 0.95$; and if the Geneva wheel is integral with the spindle carrier, drum, etc., i.e., if the diameter of the bearing surface is larger than the outside diameter of the wheel, $\eta \cong 0.75$.

Flat Geneva Wheel Mechanisms with Internal Engagement

Geneva wheel mechanisms with internal engagement are less often used in machine tools than those with external engagement. The principal kinematic and dynamic relations are derived similarly for both types.

Employing the same notation as on pages 161 to 165, the following kinematic relationships are obtained for Geneva wheels with internal engagement:

$$\frac{t_i}{T}=\frac{z+2}{2z} \text{ and } \frac{t_r}{T}=\frac{z-2}{2z} \tag{151a}$$

or

$$\left.\begin{aligned} t_i &= \frac{z+2}{z}\,\frac{30}{n}\ \text{sec} \\ t_r &= \frac{z-2}{z}\,\frac{30}{n}\ \text{sec} \end{aligned}\right\} \tag{152a}$$

and the working time coefficient of the Geneva wheel is

$$k=\frac{z+2}{z-2}>1 \tag{153a}$$

We can write for a random position of a Geneva wheel mechanism with internal engagement:

$$\tan\psi=\frac{\lambda\sin\varphi}{1+\lambda\cos\varphi} \tag{155a}$$

Hence, the angular velocity ω_w and angular acceleration ε_w of the wheel are

$$\omega_w=\frac{d\psi}{dt}=\frac{\lambda(\cos\varphi+\lambda)}{1+2\lambda\cos\varphi+\lambda^2}\,\omega \tag{156a}$$

$$\varepsilon_w=\pm\frac{\lambda(1-\lambda^2)\sin\varphi}{(1+2\lambda\cos\varphi+\lambda^2)^2}\,\omega^2 \tag{157a}$$

The last three equations are derived from equations (155), (156) and (157) for a Geneva wheel mechanism with external engagement by substituting angle $(\pi-\varphi)$ for φ.

The expression for the maximum angular velocity of the wheel is found in a similar manner

$$\omega_{w\,max}=\frac{\lambda}{1+\lambda}\,\omega \tag{161a}$$

In contrast to mechanisms with external engagement, the angular acceleration of the Geneva wheel in mechanisms with internal engagement reaches its maximum value at the beginning and end of the indexing motion (Fig. 103).

The value of $\varepsilon_{w\ in,\ fin}$ is calculated from equation (157a) in which

$$\varphi=\frac{\pi(z+2)}{z} \tag{183}$$

The force relationships for mechanisms with internal engagement are determined in a manner similar to that used here for mechanisms with external engagement

The choice between Geneva wheel mechanisms with external and internal engagement depends upon the specified operating conditions of the corresponding unit of the machine being designed. For the same number of slots and the same wheel indexing time, the maximum angular acceleration of the Geneva wheel for a mechanism with internal engagement is considerably higher than for the same mechanism with external engagement. This advantage of the latter type of mechanisms is frequently the deciding factor in their selection.

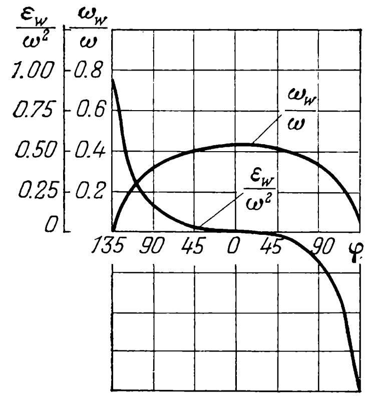


Fig. 103. Angular velocity and angular acceleration curves for a four-slot Geneva wheel with internal engagement

Spherical Geneva Wheel Mechanisms

The advantages of spherical (spatial) Geneva wheel mechanisms over the flat variety consists primarily in their capability of transmitting intermittent motion between shafts at right angles without the need of intermediate mechanisms, such as bevel gears or worm gearing, as well as their small overall size and lower inertia forces and torques. However, they have found only limited application in machine tools, mainly in semiautomatic unit-built machines and certain special machine tools (for example, in the two-way semiautomatic 8-spindle horizontal boring and thread-cutting machine, model 1C285, designed by SDO-1 [Special Designing Office, No. 1] for machining the sections of heating radiators). This is true for both Soviet and foreign makes of machine tools and can be explained by the fact that the manufacture of flat Geneva wheel mechanisms has been mastered for a comparatively long time and they have been standardized in some plants. Owing to this it is frequently more expedient to apply a flat Geneva wheel mechanism in conjunction with some other mechanical transmission than to master the production of spherical mechanisms.

7-4. Other Mechanisms for Producing Periodic Motions

Along with the mechanisms considered above for producing periodic motions, many other devices—mechanical, hydraulic, pneumatic, hydropneumatic and electrical—and combinations of them in a single or in different groups have found application in machine tools. The choice of combined

mechanisms is guided by the specified operating conditions. Thus, for example, to reduce the time required for indexing a combination of a Geneva wheel mechanism with noncircular gearing can be used; to change the transmission ratio between the driving and driven shafts, a combination of the same mechanism with circular gears can be employed, etc.

Among the mechanical devices used to produce periodic (intermittent) motions, other than cam, ratchet wheel and Geneva wheel mechanisms, are the following.

Intermittent gearing consists of a driven gear with a full tooth rim while the driving gear has teeth only over a part of its circumference. If the driving gear has z_1 teeth and the driven gear z_2 teeth, then for each full revolution of the driving gear the driven gear turns through the angle $\gamma = 2\pi \frac{z_1}{z_2}$. A transmission of this type operates with impacts at the beginning and end of the tooth engagement. If a pair of specially profiled rolling levers is added to this gearing, it can be made to operate without impacts.

Gear and worm drives powered from an individual electric motor can be applied in various ways.

CHAPTER 8

REVERSING DEVICES

8-1. Reversing Motions in Machine Tools

In the operation of most types of machine tools, a more or less frequent reversal of at least certain motions is required. This is due primarily to the fact that in most machine tools either the main drive motion or certain feed motions are rectilinear as, in most cases, are the positioning motions. In planers, shapers and slotters, both the cutting and feed motions are rectilinear. Obviously, a unit travelling in a straight line must be reversed at the end of its stroke.

In the design of certain machine tools it is necessary to make provision for the reversal of certain rotary motions as well, to enable the machines to perform all the operations they are intended for. The inclusion of a reversing device complicates the construction of the machine tool and its control and, in some cases, its manufacture or electrical circuit. Hence, reversing devices should be incorporated only in kinematic chains in which the need for them results from the purpose and functions of the chain.

In some machine tools other motions may be reversed as well. This may be due to the nature of the operation being performed or the introduction of supplementary motions which are necessary or desirable in order to obtain a better surface finish, longer tool life, etc.

All positioning motions used for setting up the machine tool or during its operation should also be reversible.

8-2. Requirements Made to Reversing Devices

Applicability Criteria for Various Reversing Systems

Motions in machine tools can be reversed by means of electrical or fluid power devices, by using purely mechanical devices or by various combinations of these. The choice of a suitable variant is governed by the requirements made to the reversing device, on the one hand, and the degree to which these requirements are complied with by various reversing systems or versions of them. As usual, in comparing variants that are equivalent as to performance, producibility and economic considerations become the deciding factors.

Regardless of the system used and its construction, a reversing device should satisfy the following main requirements:

(a) it should be capable of transmitting torques of the maximum required magnitude in each of the directions of motion, these torques frequently being inequal;

(b) inertia forces, acting during reversal, should not lead to premature wear of the components of the reversing device;

(c) the loss of energy due to reversing should be as small as possible, especially in case of frequent reversals;

(d) the overall size of the reversing device should be sufficiently small;

(e) if the reversing device does not operate automatically, the force required to actuate it should be the smaller, the more frequently reversing is to be effected, and should in no case lead to operator fatigue.

Other requirements made to reversing devices concern the frequency of reversals, time required for each reversal and the accuracy of reversing as to time and place. The strictness of these requirements depends upon the function of the reversing device in the machine tool being designed. Electrical, fluid power and mechanical solutions satisfy these requirements to different degrees.

Up to the present time, electrical reversing has found especially extensive application in machine tools. It is accomplished, for instance, by means of a shunt-wound d-c motor, whose advantages include not only intensive braking during reversal, but the possibility of obtaining infinitely variable speeds for the corresponding unit of the machine tool. However, this reversing method also has its drawbacks (availability of a d-c power supply in the shop, comparatively high energy losses in each reversal, etc.).

The application of a motor-generator set is to be preferred since the cost of reversing is much less than with a drive powered by a shunt-wound d-c electric motor.

The problems of electrical reversing in machine tools are considered in detail in special literature.

Electrical reversing deserves preference in the drives of positioning motions, for traversing heavy units of large machine tools (including heavy cluster gears in the mechanisms of heavy unique machine tools) and in mechanized devices for clamping various parts of machine tools. In such cases, hand traverse is a source of intensive operator fatigue, the time lost in reversal is not very important, settings are not made very frequently and the final positioning accuracy, if it exceeds that guaranteed by the motor, can be attained by some supplementary device, for instance, inching (jogging) controls or hand traverse of some part of the unit. In other cases, it is necessary to compare the features of the various design versions, evaluating each reversing device together with its controls. A reversible electric motor often proves to be the most convenient and economical. If different forward

and reverse speeds are required, a two- or multiple-speed motor can be employed.

The high frequency and rapidity of reversals obtained with a hydraulic drive are not yet practically attainable with a drive powered by a reversible electric motor. This is explained by the fact that in the latter case, in each reversing process it is necessary first to absorb the kinetic energy of the massive rotor, running at a high angular velocity, and then to accelerate the rotor again to the same or different (but also high) velocity in the opposite direction. At the same time, parts of the unit being reversed are also braked and accelerated. In a planer, for instance, such parts include the gears in the train to the table rack, their shafts and the table with the workpiece. Of decisive importance is the rotor of the motor constituting from 80 to 95 per cent of the kinetic energy of the reversed mass.

Conditions are more favourable in hydraulic reversing; the hydraulic circuit contains no rotary reciprocating components possessing high kinetic energy at the moment when reversing begins. No rotary reciprocating gearing, running at more or less high speeds, in the train to the reversed unit, is used with a hydraulic drive. In addition to the piston, only components of the valve gearing, small in diameter and light in weight, are periodically reversed in the circuit. These components, such as valve spools and plungers, are shifted from a state of rest and, consequently, their movement requires very little time.

Because of the comparatively small inertia forces, the accuracy of reversal achieved with hydraulic reversing is very high and depends mainly upon the inertia of the reversed mass of the machine tool itself (hydraulic reversing devices are taken up in Vol. 2, Part Four).

Notwithstanding their great advantages, electrical and hydraulic devices cannot be used for reversing in all cases or in all machine tools. A number of conditions limits the application of reversible electric motors, while hydraulic reversing proves convenient only in machine tools in which the main (working) motions are powered by a hydraulic drive.

The frequency of reversal feasible with a mechanical device can be very high and is restricted only by the inertia forces of the reversed mass.

The time τ and the accuracy of reversal depend upon the same factors as the frequency. If the kinematic train of the unit being reversed includes elements or transmissions that permit slipping, for example friction clutches or belt drives, or flexible links, the time τ will be greater than without such elements in the train. Slipping leads to a reduction in impacts, and reversals ares moother. The time τ is extremely short for reversing mechanisms having only rigid connections between the links. As the clearances in the mechanism increase due to wear and are not automatically compensated for (for instance, as in articulated joints or gearing) the reversal time increases and the reversing process is accompanied by impacts in the mating part

having excess clearance. The reversal accuracy, as to time and place, depends to a great degree upon the number and magnitude of these clearances.

The most important performance characteristics of a reversing device (permissible frequency of reversing, reversal time and reversing accuracy) can be improved by a more expedient design in which provision is made for elements which eliminate backlash and other clearances and reduce the inertia forces acting during reversal. This is sometimes achieved by making components, whose kinetic energy is of prime importance in reversal, of light alloys or of welded construction.

8-3. Energy Losses in Reversals

The process of reversing from some angular velocity ω_2 to velocity ω_1 in the opposite direction of rotation (or from a linear velocity v_2 to the reverse velocity v_1) consists of two phases: braking from ω_2 to 0 and acceleration in the reverse direction from 0 to ω_1 (or in a similar manner for rectilinear motion). Generally speaking, the velocity does not vary during reversals according to a linear dependence as has been assumed in the curve of Fig. 104 in which the acceleration is constant during the reversing periods.

The problem of reducing the energy loss and prolonging the service life of the components of the reversing device acquires great importance if some unit is periodically reversed at high frequency (for example, the table of a planer). An analysis shows that the braking of a shaft being reversed by means of a clutch rotating at the same speed in the opposite direction results in a loss in energy threefold that in stopping the shaft with a separate brake. Consequently, in order to reduce energy losses and to increase the service life of the friction parts or clutch linings, it is more advantageous to design the reversing mechanism in such a manner that the kinetic energy is absorbed by a brake and the clutch is used only for acceleration in the opposite direction. If the loss in energy in acceleration is also taken into considera-

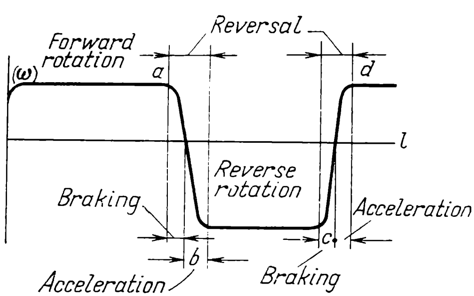


Fig. 104. Velocity variation diagram of a reversed unit

tion, it turns out that for the case in which the forward and reverse velocities are equal, the total energy loss during the whole reversal period is only one half as much when braking is effected with a brake and not with the clutch.

The clutches and brake of a reversing mechanism should have controls with an interlocking feature which makes it impossible to switch over from one clutch to the other without applying the brake between the two clutch engagements. This is most simply accomplished by single-lever (or single-button) control of the reversing device.

Since reversing devices with clutches, including those with electromagnetic clutches, are being superseded by other devices to a greater and greater extent, these mechanisms will not be considered here in more detail.

8-4. Constructions of Reversing Mechanisms

Spur and bevel tumbler units, planetary and worm gearing are used as elementary reversing mechanisms in machine tools. Belt reversing drives, which found fairly wide application previously, are very rarely used in modern machine tools and therefore will not be described in the following, along with chain drives used for the same purpose in a few models.

If the reciprocating rectilinear or rotary motion is accomplished by a slider-crank, link or cam mechanism, there is no need for a special reversing device.

Spur and helical idler gear reversing mechanisms. Idler gear mechanisms made up of spur or helical gears are extensively employed for reversing a shaft parallel to the driving shaft. Motion is reversed by transmitting it through an even or odd number of idler gears, most often through one idler gear for one direction of rotation and by direct engagement of the gears on the driving and driven shafts or through two idler gears for the other direction of rotation.

The most frequently used versions of idler gear reversing mechanisms have sliding gears or sliding double clusters of identical gears, gears in constant mesh and engaged by clutches or a sliding key, and tumbler gears which are brought into engagement by swivelling them about a stationary axis (tumbler gear reversing unit).

Diagrams of the first type of idler gear reversing devices are illustrated in Fig. 105*a* through *e*. The idler gear is denoted by z_0 in all cases; the dashed lines show the position of the sliding gear after reversal. The minimum constructional length of the unit, expressed as a multiple of the face width b of an ordinary gear, and the gearing ratios for forward and reverse rotation are given for each diagram. Figures *I* and *II* designate the driving shaft run-

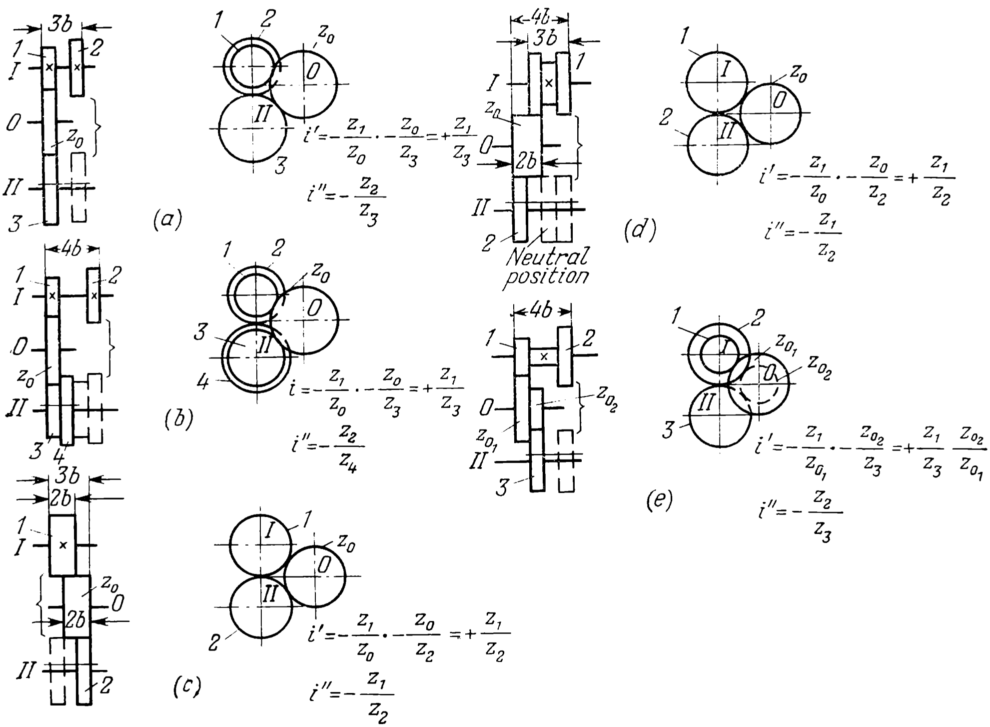


Fig. 105. Idler gear reversing mechanisms with sliding gears

ning in a constant direction and the reversible driven shaft, respectively. All the arrangements shown in Fig. 105 can be inverted so that shafts *I* and *II* are interchanged.

Reversing devices of the types shown in Fig. 105 are very simple in construction. Gears mounted on a single shaft (for instance, gears *1* and *2* in diagrams of Fig. 105*a* and *b*) are often designed as a double cluster gear; sliding gears and cluster gears are mounted on spline shafts. Bushings are press-fitted into idler gears, that run freely on their axles, to prolong their service life.

If, in the mechanisms in Fig. 105, each sliding gear is replaced by two gears which are constantly in mesh with their mating gears and are engaged to their shaft by means of clutches, we will obtain the second type of idler gear reversing devices. Diagrams of the most commonly used devices of this type are shown in Fig. 106*a*, *b* and *c*. They are obtained from diagrams (Fig. 105*a*, *c* and *e*) when these are converted as indicated above.

Jaw clutches and, less frequently, gear clutches are used to switch over the reversing units in the feed mechanisms of lathes, vertical turret lathes and milling machines. In the headstocks (or speed gearboxes) of such machines, requiring frequent reversals, a reversing device with two friction clutches is used for forward and reverse spindle rotation, if a reversible electric motor (see above) is not used for this purpose. The most extensively used arrangements of these mechanisms, whose principle of operation requires no explanation, are illustrated in Fig. 107*a*, *b* and *c*. Constructions incorporating these principles are shown in Figs. 32 and 35. It is good practice to make provision for lubricant supply from inside the clutches to reduce heating and wear of the friction surfaces from slippage. The arrangements shown in Fig. 107*b* and *c* are used when the number of speed steps in forward rotation should be more than in reverse rotation (or vice versa).

The moments of inertia of multiple-disk friction clutches are, as a rule, much higher than those of gears. Hence, it follows that it is more advantageous to arrange such clutches on the driving shaft as shown in the diagrams of Fig. 107. If the clutches are mounted on the driven shaft, the gears of the reversing mechanism will be in operation even when the clutches are disengaged. The moments of inertia of jaw clutches are less than those of gears and such clutches are mounted on the reversible shaft.

Bevel gear reversing devices, consisting of bevel gears, are used in various types of machine tools, in working feed and rapid traverse mechanisms, in roll mechanisms, etc. The main advantage of a bevel gear reversing device is that it is equally applicable for any relative positions of the driving and driven shafts. Its drawbacks are the comparatively large overall size in transmitting high torques and noisier operation when compared to spur idler gear mechanisms. As can be seen in Fig. 108, in which *I* and *II* are the driving and driven shafts, respectively (the former rotating in a constant direction and the latter being reversible), these shafts may be coaxial (Fig. 108*a*), parallel to each other (the chain line in the same diagram), square to each other (Fig. 108*b*), or arranged at an angle not equal to 90° (Fig. 108*c*). At the same angular velocity of the driving shaft, the angular velocity of the driven shaft can be the same in both directions (Fig. 108*a* and *b*) or different (Fig. 108*c* through *f*). In the latter case, however, the construction of the mechanism is more complicated.

In the same manner as spur idler gear reversing devices, the bevel gear type can also be inverted in the sense that either shaft *I* or *II* can be the reversible one.

If the reversible shaft runs at low speed, it practically does not matter on which of the two shafts the keyed gears are mounted. Otherwise, the idler gears and the clutch should be arranged, whenever possible, on the reversible shaft.

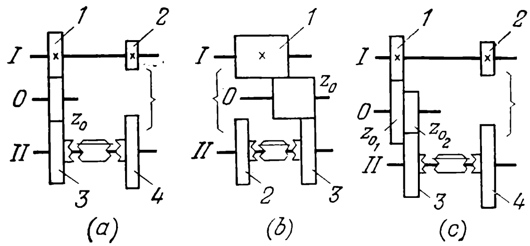


Fig. 106. Idler gear reversing mechanisms with positive clutches

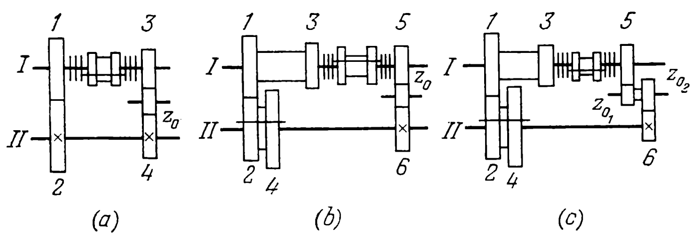


Fig. 107. Idler gear reversing mechanisms with friction clutches

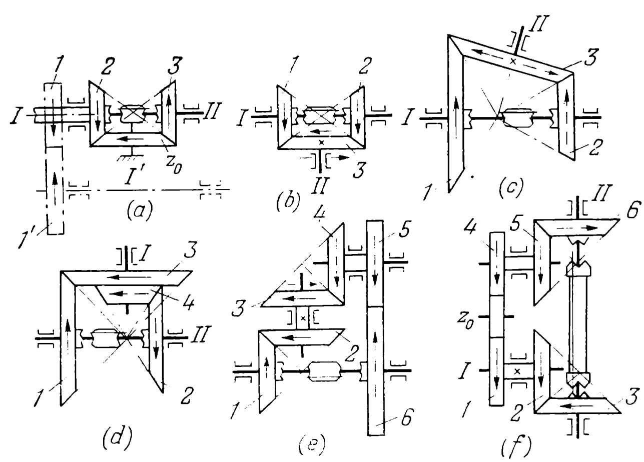


Fig. 108. Bevel gear reversing devices

Rotation is reversed in these devices either by the engagement of jaw or friction clutches or by shifting bevel cluster gears. Electromagnetic clutches have been successfully used for this purpose in machine tools for high-velocity machining.

As to its manufacture, a bevel gear reversing device is more complex than a spur idler gear reversing mechanism. This has led in recent years to the replacing of the bevel gear type by the simpler and cheaper spur (or helical) idler gear type, wherever possible, retaining the bevel gear type only for reversing shafts whose axis is square to the axis of the driving shaft.

Planetary gear reversing mechanisms enable not only the reversal of rotation, but any transmission ratios—both very large and very small—to be obtained. This makes them especially suitable for the feed trains of machine tools where a large reduction is required to effect working feeds in conjunction with rapid return traverse.

In comparing a version with a planetary gear reversing device with other mechanisms for the same purpose, it is necessary to take into account the loss of energy in a planetary mechanism which depends upon the arrangement and quality of manufacture and may be quite high. The low efficiency of the mechanism is of no practical importance if the absolute power transmitted by the drive is low. Consideration should also be given to the fact that the assembly of planetary mechanisms sometimes requires quite a high labour input.

Worm gearing reversing mechanisms find only limited application in modern machine tools. The principle of such devices is clear from the diagram in Fig. 109 in which figure *I* denotes the driving shaft and figure *II*, the driven reversible shaft. The controls of clutches *a* and *b* must be designed with an interlocking feature excluding their simultaneous engagement.

Certain models of gear-cutting machines incorporate geared mechanisms which convert rotary motion of gear *1* (Figs. 110 and 111) into rectilinear reciprocating motion (Fig. 110) or rotary reciprocating motion (Fig. 111) of unit *2* of the machine which may be a ram, slide, table, cradle, etc.

The driving element of these mechanisms is a gear rotating, as a rule, at constant speed in one direction. The driven element is a pair of parallel gear racks connected at their ends by two half-gears, in the first type of these mechanisms called the Napier motion (Fig. 110) or two concentric segment gears connected at their ends by the same half-gears (Fig. 111). In both cases, meshing may be either internal (Figs. 110*a* and 111*a*) or external (Figs. 110*b* and 111*b*).

At constant speed of the driving gear, the reversing mechanism in Fig. 110 provides uniform rectilinear reciprocating motion of the driven unit of the machine over the sections corresponding to meshing of the gear with

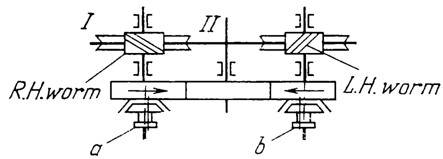


Fig. 109. Worm gearing reversing mechanism

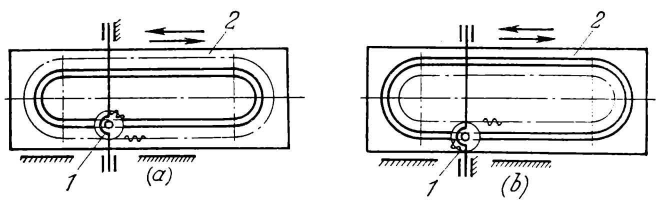


Fig. 110. Diagram of a geared reversing mechanism for converting rotary motion into rectilinear reciprocating motion

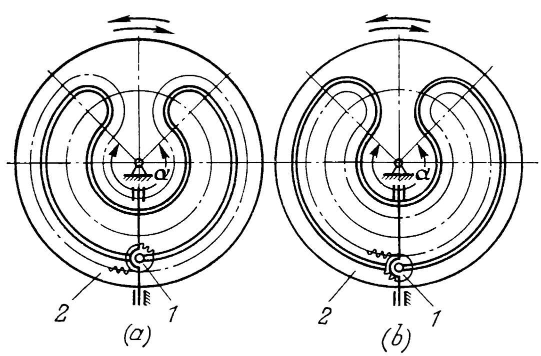


Fig. 111. Diagram of a geared reversing mechanism for converting rotary motion into rotary reciprocating motion

the racks, and reversal in the sections of path corresponding to meshing of the gear with the end half-gears.

The mechanism shown in Fig. 111 provides uniform rotation of the driven unit of the machine (though at different speeds in each direction) over the sections corresponding to meshing of the constant-speed driving gear with the segment gears (whose centre angle is α) and reversal of this unit at the sections corresponding to meshing of the driving gear with the end half-gears of the composite gear unit.

In the first type of mechanisms (Fig. 110) the velocity of the driven unit varies during reversal according to a cosine curve; in the second type (Fig. 111), the velocity varies smoothly between the uniform velocities of the forward and reverse rotation of the unit.

The reversing mechanism shown in Fig. 111 has been used, for example, in the train for cradle roll and blank indexing in the spiral bevel gear generators, models 525 and 528.

The formulas for kinematic calculations of geared reversing mechanisms of these types were derived by N. Niburg and can be found in the *Machine-Building Handbook*, Vol. I (Moscow, 1960). The method used in designing ordinary gearing is valid for dynamic calculations.

In designing reversing mechanisms, it is advisable to take into consideration the variation of the forces acting during reversals. This involves the introduction of a service life and variable-duty factor into the calculations according to the general method developed by D. Reshetov.

If these values cannot be estimated with sufficient accuracy in designing a new machine tool, it will be necessary to carry out calculations with a certain margin, basing them on the maximum acting forces, mainly the inertia forces in the reversal period.

CHAPTER 9

BEDS, COLUMNS, TABLES, CROSSRAILS AND CARRIAGES

9-1. Beds, Bases and Columns

The main requirement made to the bed, base or column of a machine tool is that it maintains the proper relative positions of the units and parts mounted on it over a long period of service under all the specified working conditions. This is achieved by designing locating datum surfaces on the bed, base or column for the principal units whose positions remain unchanged under the above-mentioned conditions. The locating datum surfaces for the travelling or adjustable units and parts are straight-line bearings called ways or guides, or ways along which the unit can be adjusted.

It follows that, along with the requirements of strength, producibility, low metal requirement and sufficiently low cost, the most important requirement made to beds, bases and columns is shape invariability. This property depends upon: (1) proper selection of the bed material and the manufacturing process, (2) the provision for a static and dynamic rigidity at which the deformation of the bed, under the action of the maximum forces during operation, is within limits conforming with the machining tolerances, and (3) a sufficiently high wear-resistance of the ways.

The configuration of a bed (base, column, etc.) is determined primarily: (1) by the arrangement of the ways on it for various units of the machine tool, (2) by the weight, dimensions and length of stroke of the main units and parts, (3) by the necessity of housing various mechanisms inside the bed, and (4) by the necessity of providing various openings, apertures, etc., in the bed walls for assembly, disassembly, inspection, adjustment and lubrication of various mechanisms of the machine, and pads, brackets and lugs on the bed walls for mounting various devices.

The operation of high-production machine tools often involves the removal of a large amount of chips, sometimes hundreds of kilograms per hour. The requirement of rapid chip disposal, one of the vital problems in designing up-to-date high-speed machine tools, materially affects the configuration of the bed which should have various openings allowing the chips to fall away freely, sloping members or chutes, etc. An example of bed construction in which proper consideration has been given to this requirement is illustrated in Fig. 112 (semiautomatic multiple-tool lathe, model 1722, made by the Ordjonikidze Plant in Moscow). Provision is frequently made in

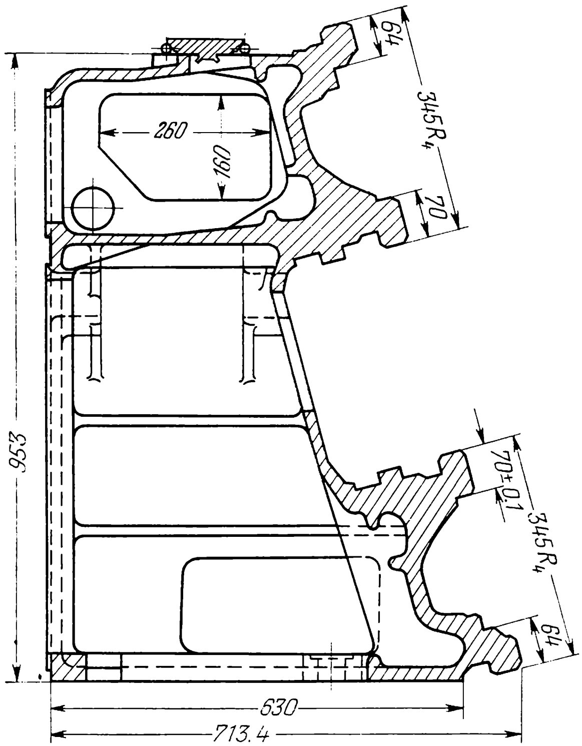


Fig. 112. Cross section of the bed of the model 1722 semiautomatic multiple-tool lathe

the beds of modern high-production machine tools for a built-in chip conveyer of the screw or other types that continuously disposes of the chips during operation.

In designing a cast bed, general foundry requirements should be complied with. Their aim is to facilitate moulding and to reduce shrinkage stresses.

As mentioned above, a bed must be sufficiently rigid. However, this is inadequate to ensure the rigidity of the machine-tool-fixture-workpiece complex. The selection of feed and depth of cut that are permissible for the required machining accuracy, the class of surface finish obtained and the specified tool life depend upon the rigidity of the whole afore-mentioned complex. This has led to the tendency to tie the main parts of the machine tool together so as to form a closed frame (Fig. 113*a* shows the open construction and Fig. 113*b*. the closed, or frame. construction), to cast the bed integral

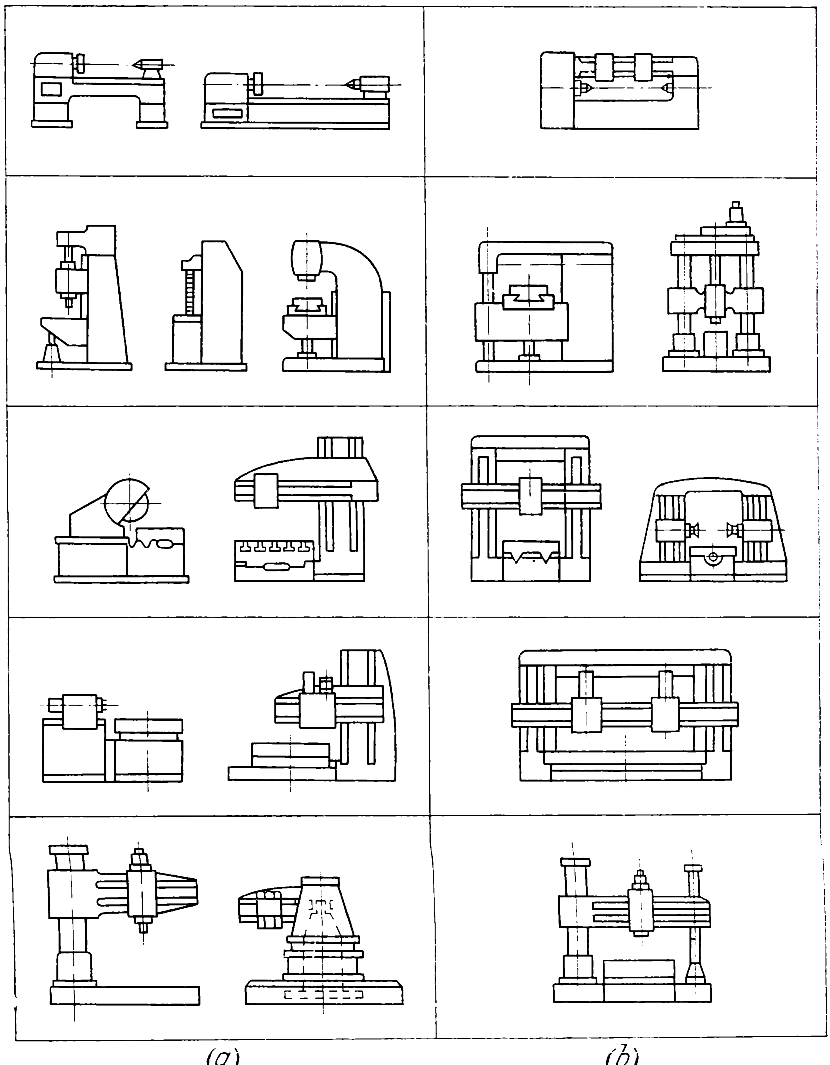


Fig. 113. Open (*a*) and closed (frame) constructions (*b*) of machine tools

with the headstock housing, and to employ a monoblock (monolithic) construction.

Ribs connecting the bed walls or cast onto the walls greatly affect the rigidity. The efficiency of ribs in increasing the rigidity of the construction depends upon their arrangement, quantity, shape and size (see below). Experiments conducted in ENIMS by H. Enikeyev on beds of various configuration showed, for example, that the arrangement of ribs (partitions) connecting the bed walls has practically no effect on the vertical rigidity of the bed. To increase the vertical rigidity, it proves expedient to cast ribs to the walls in the form of longitudinal horizontal shelves or a diagonal network. Later experimental investigations, conducted in the USSR and in other countries, confirmed the conclusions, in the main, drawn from Enikeyev's experiments and enabled additional data to be obtained for a well-founded choice of the ribbing in designing beds and other housing-type parts of machine tools.

The arrangement of the ribs and their shape have a pronounced effect on the horizontal rigidity of a bed. The most advantageous are diagonal ribs (partitions) and, in certain cases, crossed ties between the longitudinal walls of the bed. Ribbing in the form of horizontal shelves or a diagonal network has a favourable effect on the horizontal rigidity, especially if it is combined with partitions. Diagonal ribbing increases the torsional rigidity of the bed as well.

9-2. Materials for Beds, Bases and Columns

Grey cast iron. In the majority of cases, beds are made of ordinary grey cast iron, though cast irons of other types are being used to a greater extent (see below). If the ways for the travel of the units are to be cast integral with the bed (base, column, etc.), they are the deciding factor in selecting the grade of cast iron since they must possess high resistance to abrasion (abrasive wear). Most frequently employed in the USSR is cast iron with lamellar graphite, grades from СЧ21-40 through СЧ35-56, and sometimes СЧ38-60, according to USSR Std GOST 1412-54. In especially critical cases, high-strength nodular graphite cast iron (ВЧ), according to GOST 7293-54, is used. Cast iron, grade СЧ21-40, with a pearlitic matrix, is recommended for medium-size beds of not too complex shape with a wall thickness of 10 to 30 mm, and grade СЧ28-40 for a wall thickness of 20 to 60 mm. High-strength, wear-resistant cast iron, grade СЧ32-52, with a pearlitic structure, or grade СЧ35-56, can be advisably used for heavily loaded beds with a wall thickness of over 20 mm. Cast iron, grade СЧ38-60, is recommended for beds with attached ways, as well as for very heavily loaded beds with the thickest walls.

Steel. The tendency can be observed in modern machine tool engineering to replace cast beds by beds welded of rolled steel. This is due to a number of engineering and economic factors.

Cast iron possesses many advantages as a material for making beds, bases or columns (the possibility of making castings of almost any shape, good machinability, lower cost in lot production, etc.). There are, however, certain drawbacks associated with the manufacture of beds by casting. They should be taken into consideration, and include: (a) longer time is required to manufacture a machine tool due to the necessity for first making a pattern and core boxes, and for aging the casting before machining and after roughing to relieve the internal stresses; (b) possible rejection of the casting, certain defects being revealed only in the process of machining; (c) the necessity of providing quite large allowances on the surfaces to be machined; (d) if the ways are cast integral with the bed, the grade of cast iron must be selected so as to comply with the requirements made to ways; (e) the aging of large castings for a prolonged period of time slows down the turnover of the working capital and increases the total value of the unfinished goods; and (f) the expenditures on pattern and core box manufacture unfavourably affect the production costs of the machine tool if it is made in small lots (in large-lot production, the influence of this factor is so insignificant that it can be neglected).

Beds welded of previously cut pieces of rolled steel are free from the above-listed drawbacks.

The ways are either welded or bolted to the bed; hence a welded bed can be made of cheap constructional carbon steels, for example, grade Ст. 3 or Ст. 4, according to USSR Std GOST 380-60.

The elastic limit and mechanical properties of steel are much higher than those of ordinary cast iron (the mechanical properties of nodular cast iron are considerably higher than those of cast iron with lamellar graphite) and therefore much less material is required for a welded steel bed than for a cast iron bed subject to the same forces and torques, if the safety margin and rigidity (i.e., and maximum permissible deformation) of the two beds are taken to be equal. For equivalent rigidity, the weight of a steel element equals about 0.5 to 0.75 that of a cast iron element, i.e., the savings of metal ranges from 50 to 25 per cent. The actual economy of metal in replacing a cast iron bed with a steel bed depends to a great extent upon the construction of the two versions.

In making a choice between cast iron and steel as the material for the bed, base or column of the machine tool being designed, it is necessary to take into account the whole complex of engineering and economical indices of each version. The version with a cast bed is often more expedient in large-lot production, while a welded steel bed is preferable when it is necessary to make one or several machine tools in a short time.

In respect to their vibration-proof properties, beds of welded steel construction are not usually inferior to cast iron beds, notwithstanding the fact that cast iron, as a material, is more capable of damping vibrations than steel. Investigations and experience show that in an assembled construction the internal friction of the material is practically a negligible quantity in comparison to the external friction due to which vibrations are damped. The vibration-proof features of a welded bed are also due, to a certain degree, to the influence of the welds.

Welded beds for machine tools can be made of plate steel of a thickness $\delta \geqslant 3$ mm. If the walls are thin ($\delta < 8$ mm), the required rigidity can be provided by a sufficiently large number of ribs in a suitable arrangement. As a result, and also because of the large number and great length of the welds, it may turn out that the same bed, but made of steel plate 10, 12 or even 15 mm thick, is not heavier and, at the same time, is simpler to make than a bed with thinner walls.

In addition to the materials mentioned above, alloyed cast irons and nitrided cast iron have found some application for making beds.

Concrete. Reinforced concrete has been used to some extent, in the USSR and other countries, for making the beds of heavy machine tools. Shown in Fig. 114 is a cross section of a reinforced concerete bed of a heavy lathe, model 1660, manufactured by the Kramatorsk Heavy Machine Tool Plant for turning work up to 1250 mm in diameter, 6300 mm long and weighing up to 30 tons. This bed was designed and manufactured in place of the ordinary cast iron bed (grade СЧ21-40) shown in Fig. 115. To estimate the rigidity of the experimental reinforced concrete bed in comparison with the cast iron bed, they were both subjected to a horizontal spreading force (6.5 metric tons) applied between the bed ways over the left partition, between the partitions and over the right partition. The total deflections of the beds at these three places were 0.26, 0.26 and 0.25 mm for the cast iron bed and 0.167, 0.135 and 0.143 mm, respectively, for the experimental reinforced concrete bed, i.e., from 36 to 45 per cent less. These experiments show that the substitution of metal beds by reinforced concrete beds may be technically expedient and economically advantageous. Such a substitution may reduce the metal required and the production costs by about 40 to 60 per cent.

The base, both housings and certain other parts were made of reinforced concrete in the heavy vertical turning and boring mills, models 1563C and 1580C, manufactured by the Kolomensk Heavy Machine Tool Plant. The bed and other basic parts of a heavy planer were also made of reinforced concrete in the same plant.

However, due to a number of reasons involving the materials (cement), equipment required to prestress the reinforcement, and the lack of experience

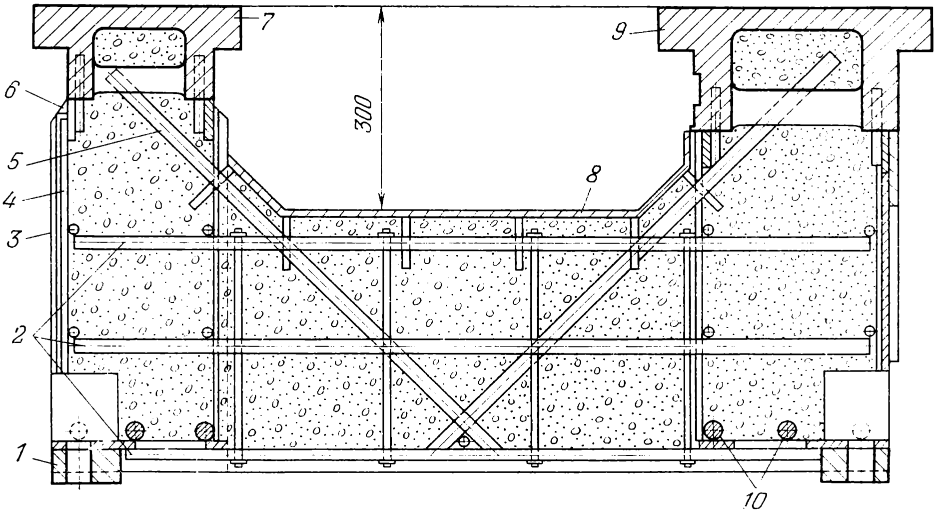


Fig. 114. Cross section of a reinforced concrete bed for the model 1660 heavy lathe

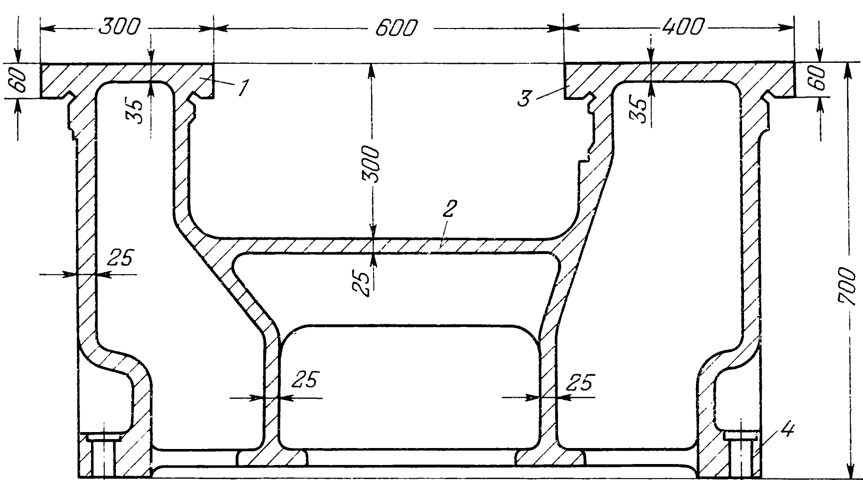


Fig. 115. Cross section (through bearing pads) of a cast iron bed for the model 1660 heavy lathe:

1 and 3—ways; 2—diaphragm; 4—bearing pad with a hole for a foundation bolt

in the corresponding machine tool plants, reinforced concrete beds (and other housing-type parts) have not yet found wide application in machine tools, even though they show promise. The main material for making beds, bases and columns is still cast iron.

9-3. Typical Constructions of Beds, Bases and Columns

The construction of a bed, base or column is based on certain general principles stemming from the following circumstances.

During machine tool operation, the cutting forces, weight of the stationary and travelling parts, weight of the workpiece, etc., and in some machine tools, inertia forces act on the bed (base, column, etc.). These forces lead to stresses and deformation in the material of the bed of a type that can be established on the basis of an analysis of the system of forces acting on the bed in the cutting process. In critical cases, especially in designing heavy machine tools, it is also necessary to analyze the periods of transitional motion—acceleration and braking—when the inertia forces and frictional resistance play an especially significant part.

It is impossible to accurately determine the deformation of the bed being designed, especially if it is of complex shape, by calculations. The required rigidity is usually provided for by methods proved in practice.

In tension and compression, the factor of safety n and the rigidity S of an element of the construction depend, all other things being equal, upon its cross-sectional area, but not upon the shape of the cross section. Consequently, in these cases, the material requirement is fully determined by the acting forces and the selected values of n and S. In bending and torsion, on the other hand, the metal requirement can be reduced by a proper selection of the shape of the cross section of the element in which the section moduli and moments of inertia are increased without changing the cross-sectional area, i.e., without changing the weight of this element of the construction.

It can be readily proved that a cross section in the form of a hollow rectangle is the most advantageous insofar as rigidity in bending, and especially in torsion, is concerned. Since producibility considerations are also in favour of this shape of cross section it is most often the basis for bed design.

It is usually impossible to retain a cross-sectional shape with a completely enclosed rectangle, or even one enclosed on three sides, over the whole length of the bed because of the necessity for ensuring free chip disposal, for arranging various mechanisms inside the bed, etc. This substantially reduces the rigidity of the bed. Therefore, to increase the rigidity of powerful machine tools, efforts are made to retain a longitudinal horizontal stiffening rib (usually of trough shape) unbroken over the full length of the bed.

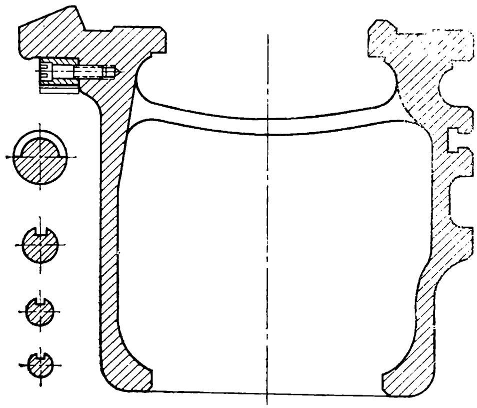


Fig. 116. Cross section of the bed for the Krasny Proletary lathe

This rib is either inclined or has openings (Fig. 116) to facilitate chip disposal.

A very effective means of attaining the required rigidity, and one applied in all beds, is the provision of partitions which may be of the transverse type, tying together the longitudinal walls, or of the less frequently employed longitudinal type.

Transverse partitions, arranged as shown in Fig. 117*a* and *b* (plan views), are extensively used in machine tool beds. The superiority of diagonal partitions over the parallel type is evident from Fig. 117 and was conclusively proved by the experiments of H. Enikeyev on cast iron beds. Diagonal partitions are widely used in up-to-date machine tools of the medium (Fig. 118) and large sizes. Parallel partitions are employed in heavy and medium-size machine tools of various types. They either have a continuous cross section or they are of hollow inverted-U (arched) shape (Fig. 119). Frequently, a bed is strengthened by a combined system of walls, partitions and stiffening ribs. Examples of such machines are the horizontal and vertical constructions shown in Figs. 120 and 121 (base and column of milling machines of the Kearney and Trecker Corp., USA).

The beds of heavy machine tools are often of sectional construction. In designing such a bed, it is necessary to provide means in the construction to obtain a sufficiently high rigidity of the joints between the sections.

The difference in the materials, requiring different manufacturing processes, does not permit the design of a steel bed to be a simple copy of cast iron machine tool beds, since with such an approach the savings in material is often insignificant and the cost is higher.

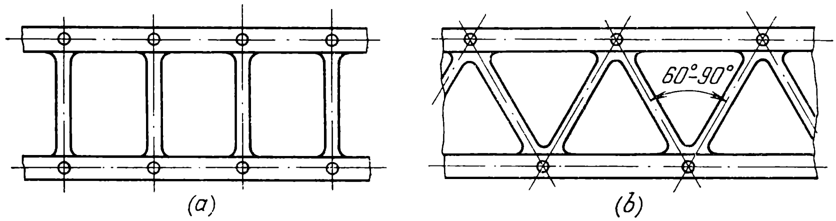


Fig. 117. Beds with parallel (*a*) and diagonal (*b*) partitions

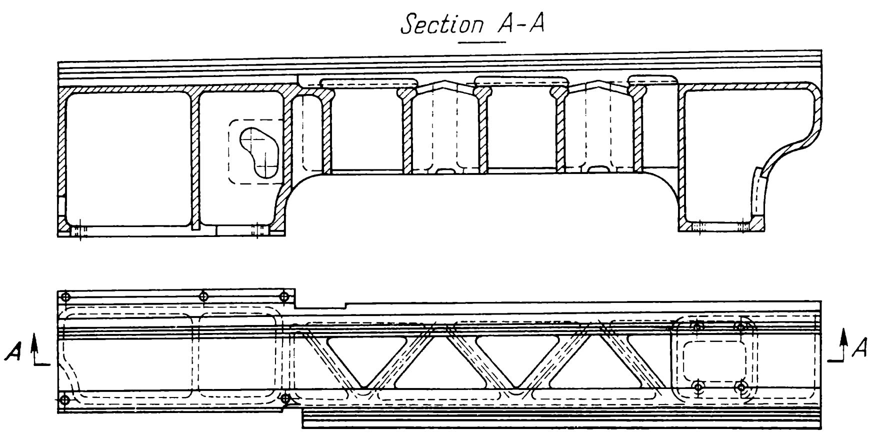


Fig. 118. Bed of the model 1Д62 engine lathe prior to modernization

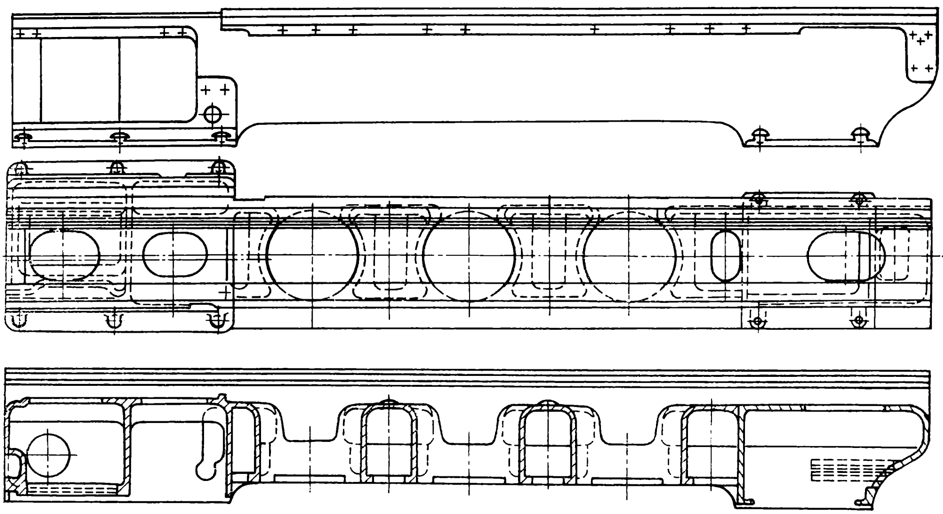


Fig. 119. Bed of an engine lathe manufactured by the Krasny Proletary Plant

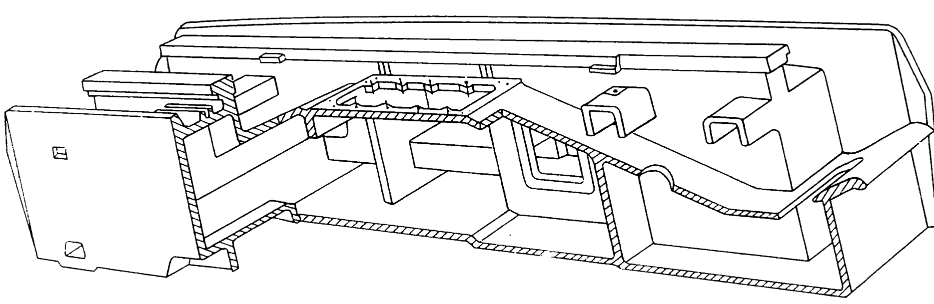


Fig. 120. Base of a milling machine manufactured by the Kearney and Trecker Corp. (USA)

Plate steel is the chief material used in making welded beds. Therefore, such beds are bounded by flat surfaces and represent a more or less complex polyhedron. The rigidity of a welded steel bed is mainly due to partitions, corner plates and other reinforcing members that tie the walls together. These members are also made of plate steel.

Thick plates are used for beds in cases when it is difficult or impossible to weld in partitions and other stiffening members. In all other cases, constructions of lighter weight can be designed of steel plate from 3 to 6 mm thick, the required rigidity being achieved by a system of suitably arranged partitions, braces and angle plates which divide the bed into a number of compartments. An example of a bed of such lightened construction, divided by partitions into compartments, is given in Fig. 122.

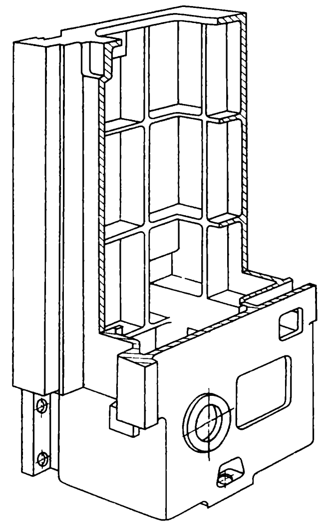


Fig. 121. Column of a milling machine manufactured by the Kearney and Trecker Corp. (USA)

The working drawing of a bed should carry all the dimensions required to make the pattern, if the bed is to be a casting, or the templates, if it is of welded steel construction.

9-4. Modern Machine Tool Bed Design

To carry out checking calculations on the designed bed, it is necessary first to draw up a design diagram, simplifying the configuration of the bed and assigning the magnitudes and directions of the acting forces. These forces include the components of the cutting force; weight of all the units mounted on the bed and that of the workpiece; forces developed in clamping the workpiece on the bed; inertia forces, if any (planers, shapers and slotters); and forces acting on the bed from the foundation. After this, beds (bases, columns, etc.) with an approximately straight axis are regarded as straight beams of variable cross section; beds with a curvilinear axis are regarded as curved beams.

Owing to the highly complex configuration of a bed (see Figs. 112, 118, 119, 120, etc.) and its variable cross section in both the transverse and longitudinal directions, such calculations are only approximate and tentative. They can, however, be used for a comparative appraisal of designed versions

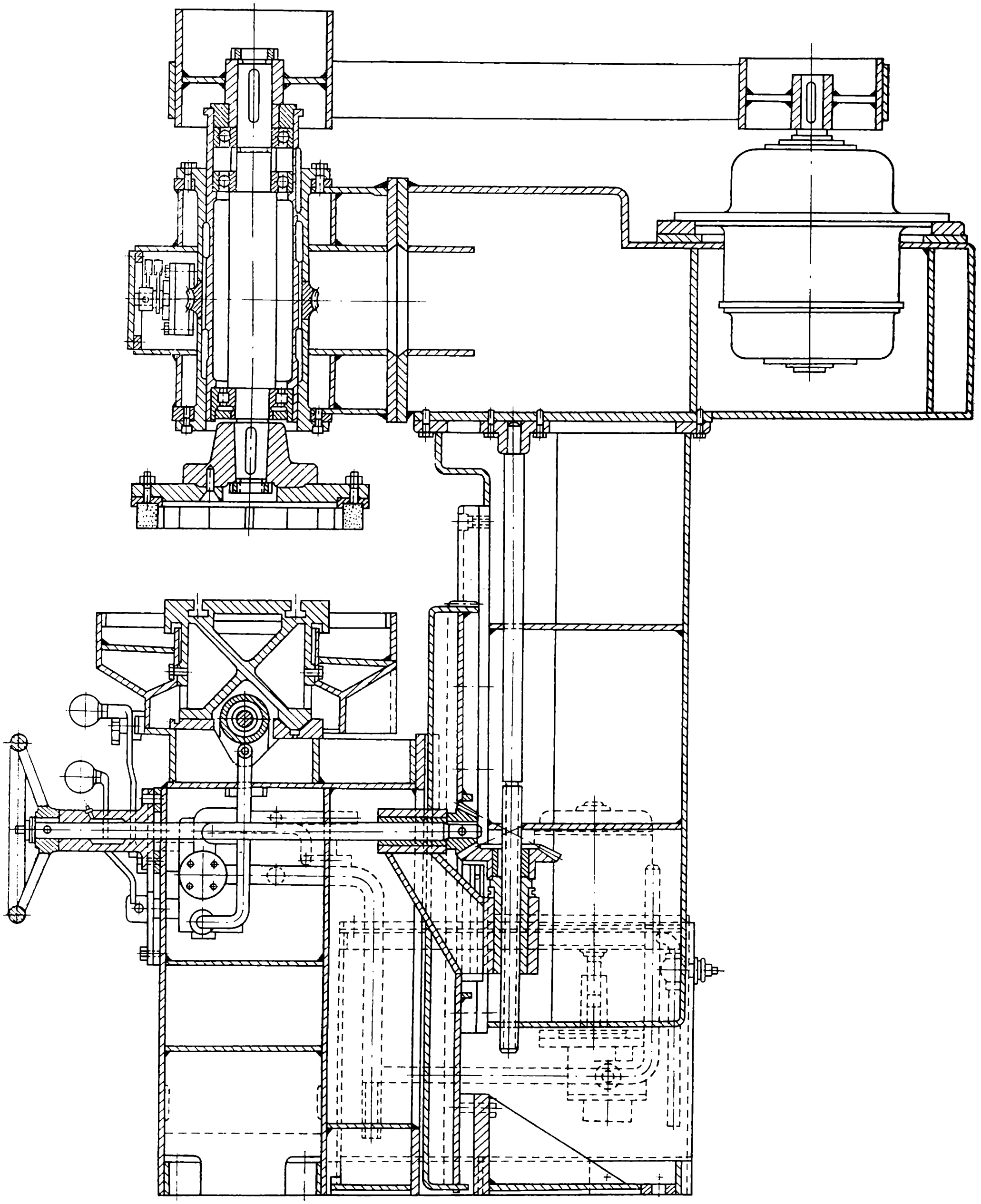


Fig. 122. Surface grinder with a welded steel base

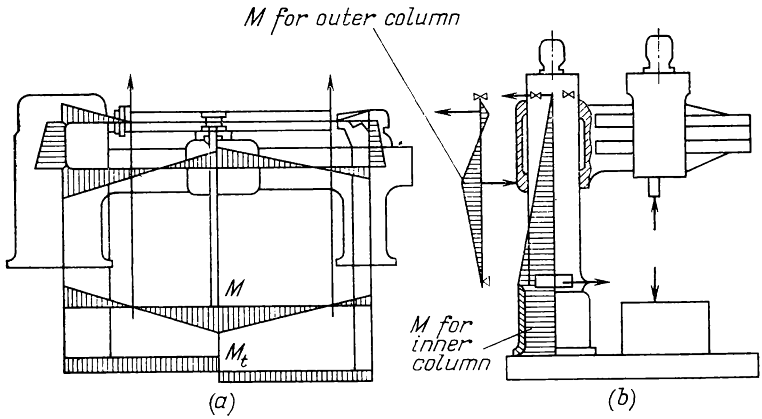


Fig. 123. **Design,** bending moment and torque diagrams for a lathe (*a*) and radial drill (*b*) (after D. Reshetov)

of the bed, as well as to estimate the order of magnitude of the stresses and bed deformation.

Design diagrams and diagrams of the bending moments M and of the torques M_t, worked out by D. Reshetov for a lathe and a radial drill, are given as an example in Fig. 123.

The method of calculating the stress due to bending is well known from the study course *Strength of Materials.* Calculations of the stress due to torsion are treated below.

No generally accepted system of rigidity indices exists. The following value is usually taken as the flexural rigidity index:

$$S = \frac{P}{f} \text{ kgf per mm (or kgf per micron)} \tag{184}$$

where P = acting force, kgf

f = resulting deformation (or movement), mm (or microns).

The rigidity depends upon the elastic properties of the material, i.e., Young's modulus of elasticity E in bending; the shape of the cross section of the beam, substituting for the bed in the calculations and, therefore, the moment of inertia I of the cross section; and the curvature $\frac{1}{\rho}$ of the beam axis, bent by the moment M. For this reason, the rigidity is sometimes taken to be

$$S = IE = M\rho \tag{185}$$

The torsional rigidity is characterized by the ratio

$$S_t = \frac{M_t}{\theta_1} \tag{186}$$

where M_t = torque

θ_1 = angle of twist per unit of length.

In this case, the torsional rigidity S_t and the flexural rigidity S are expressed in the same units.

Bed cross sections are noncircular, open for the most part, and have walls that vary around the contour. Therefore, it is impossible to calculate the angle of twist and the torsional stress to any appreciable degree of accuracy. Approximate methods are employed for calculations of this kind, replacing the actual cross sections of the bed with simpler ones that are identical along the whole length of the bed, or a scale model is investigated (see pages 25 and 26).

In calculations for beds with a closed hollow cross section and with ordinary relationships between the dimensions of the cross section and the wall thickness, the following formula can be used for a thin-walled closed contour of random shape

$$\int \tau \, ds = 2GF\theta_1 \tag{187}$$

where τ = shearing stress

ds = element of the contour

G = modulus of elasticity in shear

F = area bounded by the centre line of the walls

θ_1 = angle of twist per unit of length.

It can be assumed that the shearing stresses τ are uniformly distributed over the wall thickness δ in thin-walled closed profiles, operating with the vector $\tau\delta$ = const which is directed tangent to the middle contour of the wall in the cross section. In this case

$$\tau\delta = \text{const} = \frac{M_t}{2F} \text{ and } \tau = \frac{M_t}{2F\delta} \tag{188}$$

where M_t is the acting torque. Substituting this value of τ in equation (187), we can write

$$\int \frac{M_t \, ds}{2F\delta} = 2GF\theta_1 \text{ and } \theta_1 = \frac{M_t}{4GF^2} \int \frac{ds}{\delta} \tag{189}$$

The integral along the contour, included in the last equation, is to be replaced by the sum of the ratios $\frac{s_j}{\delta_j}$. Thus, if the cross section can be regarded as closed, the angle of twist of a bed of length l is

$$\theta \simeq \frac{M_t l}{4GF^2} \sum \frac{s_j}{\delta_j} \tag{190}$$

For a cross section in the form of a hollow rectangle with outside dimensions a and b and a constant wall thickness δ we can write

$$F=(a-\delta)(b-\delta) \text{ and } \sum \frac{s_j}{\delta_j}=\frac{2(a+b-2\delta)}{\delta}$$

Then, for this case, eguation (190) becomes

$$\theta \cong \frac{M_t (a+b-2\delta)\, l}{2G\,(a-\delta)^2 (b-\delta)^2\, \delta} \tag{191}$$

On the basis of equation (188), the average torsional stress is

$$\tau=\frac{M_t}{2F\delta}=\frac{M_t}{2\,(a-\delta)(b-\delta)\,\delta} \tag{192}$$

The angle of twist of beds with open cross sections can be calculated only approximately. If the profile consists of very narrow rectangles, the angular torsional strength is taken equal to the sum of the angular torsional strengths of the rectangles making up the profile. For a narrow rectangle with a long side s_j and short side δ_j, the linear angle of twist is

$$\theta_1=\frac{M_t}{\frac{1}{3}\, s_j \delta_j^3 G} \tag{193}$$

Using the formulated rule for straightening out an open cross section, we can write

$$\theta=\frac{3M_t l}{G \sum s_j \delta_j^3} \tag{194}$$

in which the notation is the same as in the preceding equations.

The maximum shearing stress can be calculated by the formula

$$\tau_{max}=\frac{M_t}{\frac{1}{3}\, s_j \delta_j^2}=\frac{3M_t}{s_j \delta_j^2} \tag{195}$$

(for elementary profiles of rectangular form).

The maximum shearing stress is developed at the middle of the long side of the rectangle with $\delta_j=\delta_{max}$.

Taking into consideration the approximate nature of calculations in bed design, conservative permissible stress values are assigned in the order of 80 to 120 kgf per sq cm for cast iron beds and 150 to 200 kgf per sq cm for steel beds. The calculated deformation is to be assessed on the basis of its influence on the working accuracy of the machine tool and its vibration-proof features.

The problem of working out a rigorously substantiated method of calculations in bed design still awaits a solution, as do other problems associated with machine tool rigidity. Greatest progress in this field has been achieved by the following Soviet investigators: K. Votinov (from 1930); D. Reshetov, H. Enikeyev, V. Kaminskaya, Z. Levina and others of ENIMS; and A. Sokolovsky and his co-workers of the Leningrad Polytechnical Institute. The most comprehensive and detailed material on bed design can be found in the book "Beds and Housing-Type Parts of Machine Tools (Design and Calculations)" by V. Kaminskaya, Z. Levina and D. Reshetov of ENIMS and published in Russian by Mashgiz, Moscow, 1961.

9-5. Machine Tool Columns, Housings, Tables, Crossrails and Carriages

General Instructions for Their Design

Stanchions, housings, tables, carriages, slides, crossrails, as well as such components as the knee of milling machines, columns of radial drills, columns of semiautomatic multiple-spindle vertical chucking machines, and other housing-type components are distinguished for their great variety of configurations. Their configuration depends upon with what parts of the machine tool these components mate, whether the parts are fixed or movable, the location of the components in the machine tool, the magnitude and direction of the acting forces, and other factors. Various methods are used to join these components with the base or bed of the machine tool. As an example, five possible versions of the construction arrangement of portal-type (double-housing) machine tools are shown in Fig. 124*a* through *e* (after P. Dunayev). The rigidity of these versions and their producibility are far from being equal, a circumstance which must be taken into account by the designer in choosing one of the versions. Notwithstanding the great diversity of the above-mentioned parts in respect to their purposes and their multiformity in construction, certain general characteristic features can be singled out.

The principal requirements made to the housing-type parts of machine tools concern their rigidity and vibration-proof properties. Quite often these requirements are extremely high (for instance, for the tables of thread grinders and jig boring machines, and the columns of surface grinders) since the machining accuracy of the machine tools depends upon the rigidity and vibration-proof features of these parts. Also of importance are the accuracy of the surfaces used for locating the fixture holding the workpiece or for locating the measuring devices, truing attachments, etc.; the accuracy and correct geometrical shape of the surface on which the given part is mounted;

wear-resistance of ways; ease of manufacture and the minimum possible metal requirement.

The above-mentioned parts are made of the same metals used in modern machine tool engineering for making beds (see pages 189 to 191). More or less large housing-type parts are made, along with beds, of either cast or welded design. Most of what has been said of welded steel beds (see pages 190 and 191) is also true for welded housing-type parts.

The required rigidity is attained, as in beds, by the box-shaped cross section, and the system of ribbing in cast constructions (Fig. 125) or braces, angle plates and similar stiffening members in welded steel constructions.

Typical of up-to-date machine tools are deep powerful cross sections of stanchions, columns, crossrails and like parts, in conjunction with comparatively thin walls. An expedient distribution of the metal can be established by a proper analysis of the diagram of acting forces.

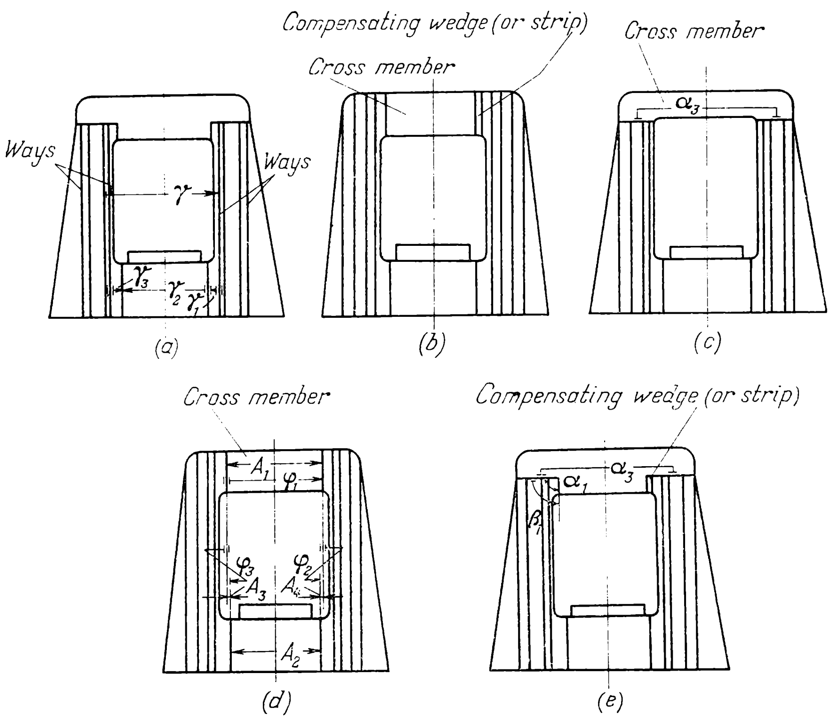


Fig. 124. Possible versions of the construction arrangement of portal-type (double-housing) machine tools

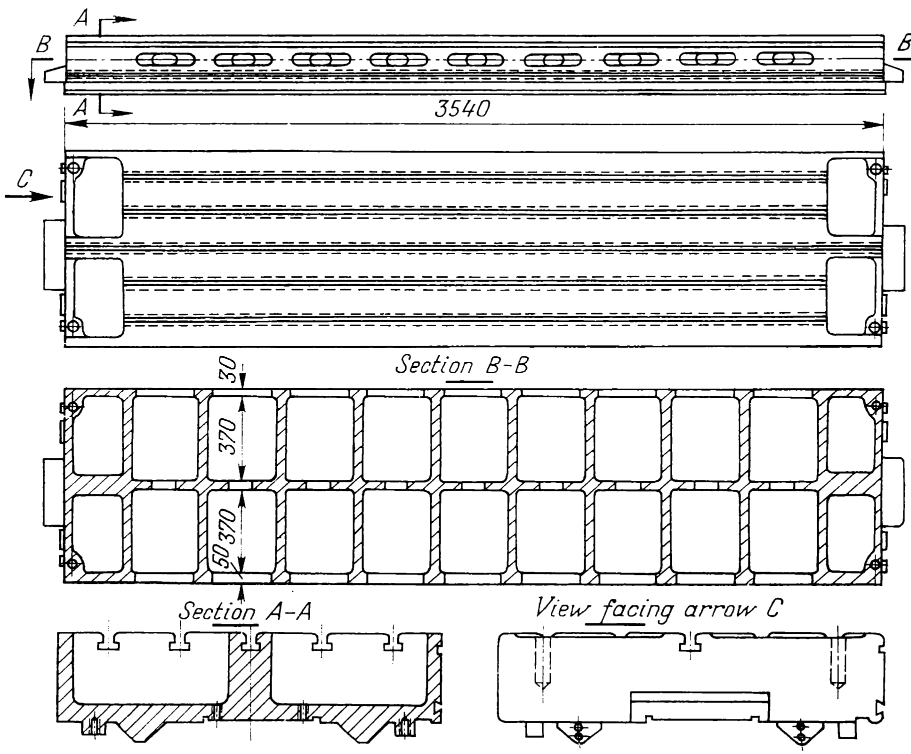


Fig. 125. Planer table

The rigidity of such parts as tables and carriages or slides depends to a great extent upon the number of joints or mating surfaces and their arrangement in respect to the acting forces. As a rule, the fewer the joints or mating surfaces, the more rigid the construction will be. However, operating conditions, in some cases, do not allow the number of joints to be reduced below a certain limit (see, for example, Figs. 126 and 127). In such cases, it is necessary, at least, to enlarge the contacting surfaces in a direction approximately perpendicular to that of the acting force, so as to reduce the specific pressure, and to make provision for firmly and reliably clamping parts which are to be stationary during operation. Similar clamping or binding devices are used to secure the outer column and arm of radial drills, crossrails of vertical boring mills, planers, planer-type milling machines, etc. These clamping devices may be hand operated or powered by an individual

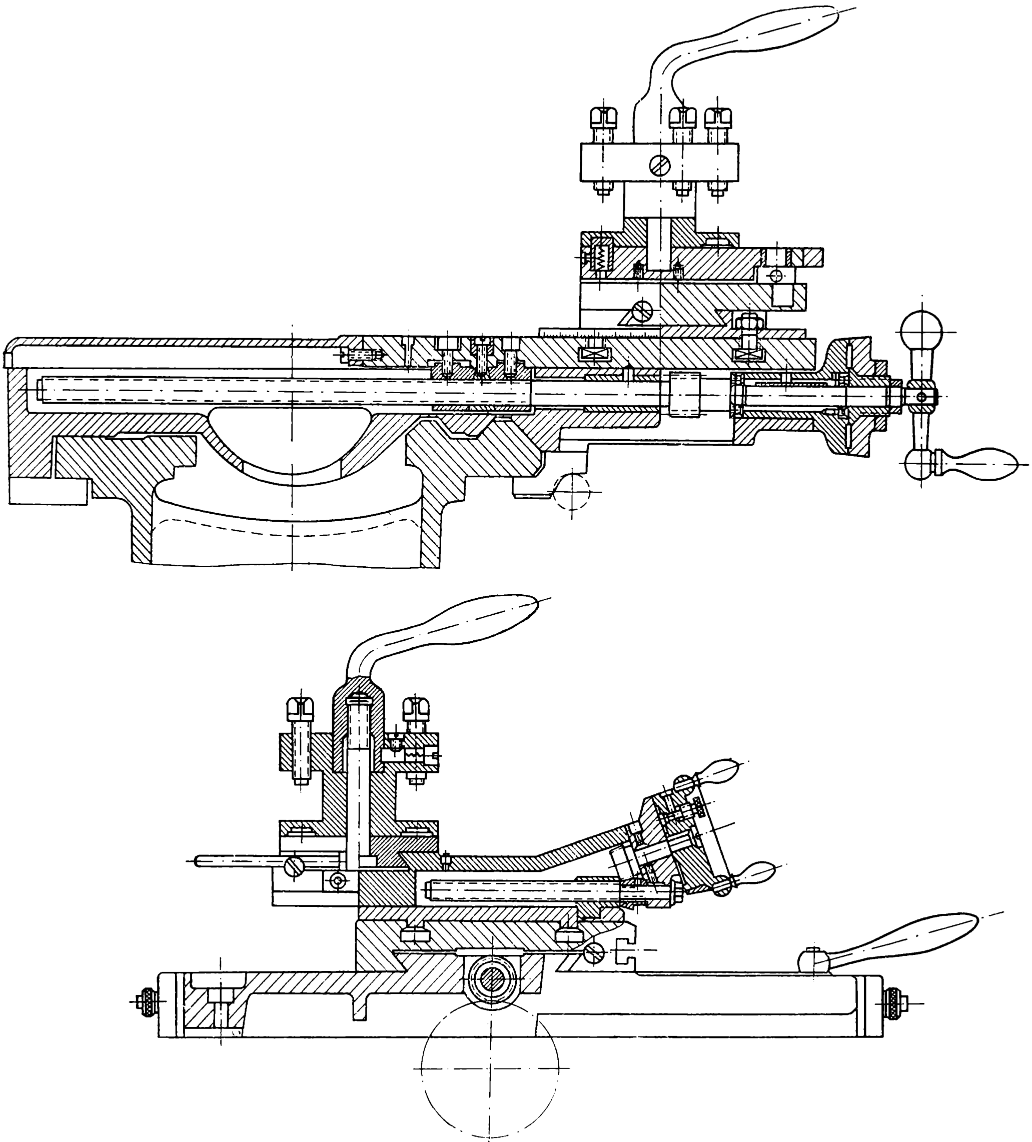


Fig. 126. Carriage of an engine lathe

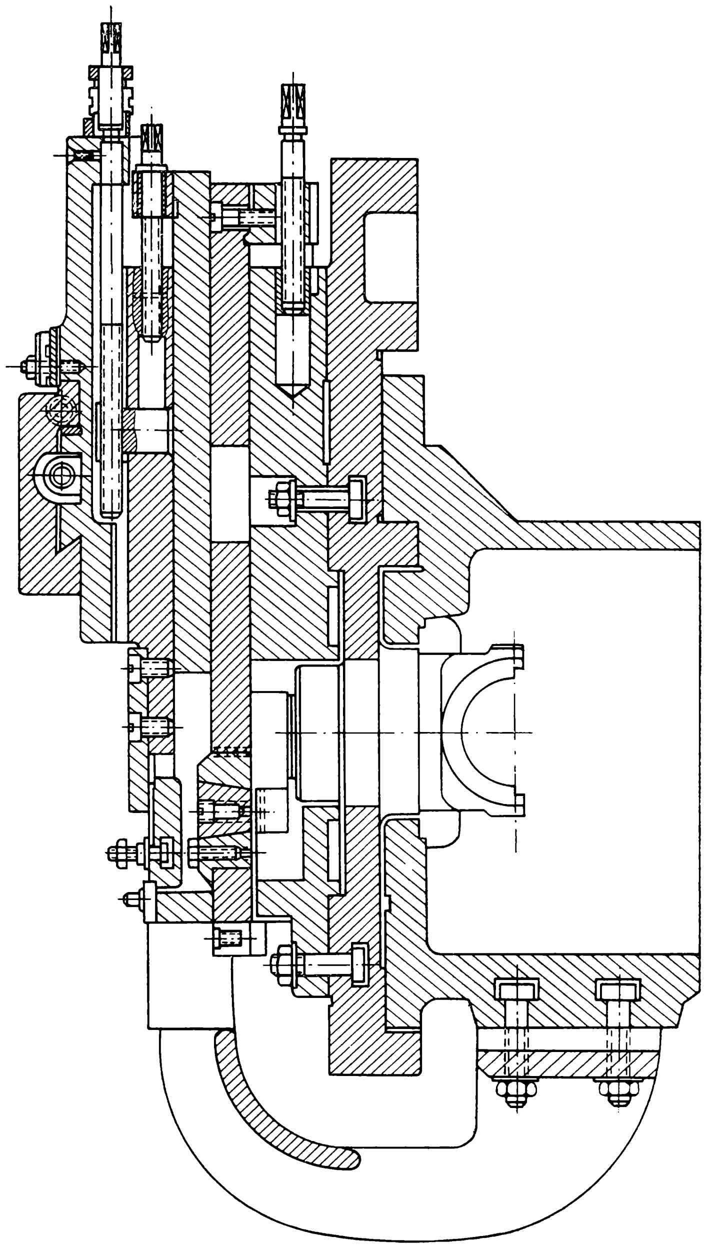


Fig. 127. Carriage of a relieving lathe

electric motor, and equipped with a device which continuously checks the firmness with which the component is clamped and switches off the main drive if it becomes loose. The condition of the clamping device is commonly indicated by coloured lamps.

As to the methods of ensuring the required rigidity of the parts under consideration, using the least amount of metal in their manufacture, all that was said on this question in respect to beds holds true (closed cross sections of definite shape, minimum possible number of openings or apertures and a reduction in their size, the use of partitions and integral ribbing, etc.). The deformation of these parts can also be reduced by the use of closed constructions in the form of frames and portals, braces, supports, etc. In precision machine tools the table should not overhang the base ways even at its extreme positions.

If a housing-type part is traversed along vertical ways by a kinematic train which contains no self-braking transmissions, the part is balanced with a counterweight or spring to facilitate its setting and to prevent it from sliding down when it is unclamped. Lubricating grooves of approximately the same type as on bed ways are made on the ways of tables, crossrails, stanchions and like parts. The horizontal working surfaces of housing-type parts are surrounded with a trough for drainage of the cutting fluid.

The working surfaces of tables have a system of parallel, and sometimes perpendicular, T-slots used to set up and clamp various types of fixtures. The dimensions of these T-slots have been standardized (USSR Std GOST 1574-62).

CHAPTER 10

WAYS

The cutting tool or the work travel in a straight line or a circle, together with the units on which the tool or work is mounted, on ways, which can also be called straight-line or circular bearings. Used widely in machine tools are *slideways* (sliding-friction ways) and *antifriction* (rolling-friction) *ways*. The latter have intermediate rolling members (balls or rollers). The principal characteristics of ways are:

1. *Accuracy of travel*, which depends mainly upon the accuracy with which the ways are machined and is characterized by the degree to which the actual travel of the unit is in compliance with strictly rectilinear (or circular) motion.
2. *Durability*, which is characterized by the capacity of the ways to retain the initial accuracy of travel of the corresponding units over a specified period of operation.
3. *Rigidity*, which is characterized by elastic displacements due to contact in the ways under the action of a normal load.

10-1. Slideways

The operating features of slideways depend both on a proper choice of material for the mating surfaces and on the construction of the ways.

Materials for Slideways

The wear of slideways depends to a considerable extent upon what materials are used to make the ways of the bed and of the travelling unit—table, saddle, slide, etc. An inexpedient selection of these materials may lead to premature wear which is not uniform along the length of the ways. This, in turn, results in an inevitable loss of accuracy of travel.

Investigations conducted by A. Pronikov established a direct relationship between the shape of the worn way and the errors in the geometric features of the work machined.

The wear resistance of slideways is determined primarily by the physico-mechanical properties of their material. A high surface hardness of ways does not, by itself, guarantee high wear resistance. Numerous experimental

data indicate that minimum total wear of slideways is attained with different hardnesses of the mating pair of surfaces due to run-in of the softer material of the pair. In most cases it is more expedient to use the harder material for the stationary slideways (bed ways) since their shape is copied in travel of the moving unit and, moreover, it is more difficult and expensive to repair the bed ways.

Grey cast iron is the most commonly used material for slideways. It is employed when the slideway is made integral with the bed and, correspondingly, with the travelling unit. The wear resistance of cast iron slideways can be increased by surface hardening performed with flame or induction heating. Such a heat treatment increases the hardness of the ways: up to 40-52R_C for ordinary grey cast iron (СЧ) and up to 45-55R_C for nodular cast iron (ВЧ).

Steel slideways in the form of strips are either welded to a steel bed, or they are secured by screws or bolts to a cast iron bed. In the USSR such steel slideways are most often made of steel 40X and then induction hardened to 52-58R_C. They are also made of steel 15 or 20X which develops a hardness of 56-62R_C after carburizing and quenching. Ball bearing steel, grade ШХ15, is also used for slideways. The use of hardened steel slideways mating with hardened cast iron ways ensures a high wear resistance.

Due to their antiscoring and anticorrosive properties, plastics are promising materials for slideways. Laminated fabric strips are used in combination with cast iron for the slideways of heavy machine tools where the comparatively low rigidity of the travelling units leads to considerable nonuniformity in the distribution of pressure on the slideway surfaces. This, in turn, may result in jamming, especially with insufficient lubrication. The drawbacks of laminated fabric ways are the low modulus of elasticity in comparison to that of steel, the tendency to swell when they absorb oil and the low coefficient of thermal conductivity. In connection with these drawbacks, it proves more advantageous to employ slideways with a thin polymeric coating applied by spraying, gluing on a thin film or some other method.

In certain cases that are justified by calculations, pads of zinc alloy, grade ЦАМ10-5, or of bronze are used on the slideways. They possess good wear resistance, but are expensive and sometimes involve the use of critical materials.

Manufacturing Specifications for Machine Tool Slideways

Specifications for the manufacture and acceptance of metal-cutting machine tools stipulate requirements in respect to the hardness, surface finish and accuracy of slideways.

The *hardness of slideways cast integral with the bed* is assigned in accordance with the standards for cast iron of the corresponding class. The permissible deviation in hardness within the limits of a single way is $\Delta Bhn \leqslant 25$ or 35, depending upon the length of the way. In case of sectional beds, the hardness deviation is $\Delta Bhn \leqslant 45$ over the whole length of the way. The Bhn value should be within the limits established for cast iron of the corresponding class.

The hardness of hardened steel slideways may be as high as $52R_C$ or even higher; nitrided slideways have a hardness of DPN 800-1000 (Vickers hardness number).

Surface finish and assembly. The ways of beds, as well as of stanchions, housings, slides, etc., should be finish machined by scraping, grinding, or any other method that produces a surface of at least the same high quality (buffing, lapping with ГОИ paste). Hardened slideways should be finished by fine grinding.

In checking the bearing contact pattern of slideways with a marking compound, the number of bearing spots in an area (25×25) sq mm should be: at least 25 for ways of precision machine tools; at least 16 for slideways of machine tools of above-standard accuracy; at least 10 for slideways with a width $b \leqslant 250$ mm and ways with a width $b \leqslant 100$ mm along which units are adjusted (but do not travel); and at least 6 for slideways with a width $b > 250$ mm and ways with a width $b > 100$ mm along which units are adjusted. The number of bearing spots is determined as the average in an area of 100 sq cm.

The degree of contact of mating slideway surfaces is checked with a marking compound and a thickness gauge 0.04 mm thick.

The *accuracy of ways* is stipulated by the accuracy standards for machine tools of various types. The required accuracy and surface finish of slideways are obtained by suitable machining techniques.

Constructions of Slideways

Rectilinear motion of a machine tool unit (table, saddle, slide, etc.) is obtained if the ways restrict free movement of the unit in all other directions. Ways which leave the travelling unit a single degree of freedom are usually called *closed ways* (Fig. 128*a*) in contrast to *open ways* (Fig. 128*b*) which are held in contact by the external load acting in a definite direction.

Closed ways can be formed of any ruled surface (except a circular cylinder) whose elements are parallel to the direction of the required motion.

The simplest of all ruled surfaces, from the point of view of manufacture and inspection, is a triangular prism with three guiding surfaces. This prism is the basis for most of the principal shapes of machine tool ways (Fig. 129*b*

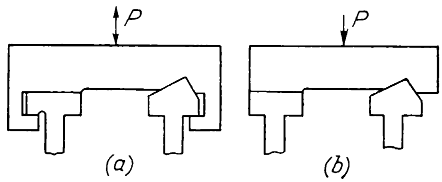


Fig. 128. Slideways:
(*a*) closed ways; (*b*) open ways

and *c*). Ways, or guides, in the form of two circular cylinders (Fig. 129*d*) are used much less frequently.

Flat ways (Fig. 129*a*) may be either vertical or horizontal. They are distinguished for the simplicity of their manufacture and of checking their geometric features. On the other hand, they require devices for adjusting the clearances, have a tendency to accumulate dirt, and retain the lubricant comparatively poorly when they are of the encompassed type.

Vee ways (Fig. 129*b*) are more difficult to manufacture than flat ways, but are capable of self-adjustment, i.e., clearances are automatically eliminated under the action of the load. The encompassed type of vee way has no tendency to accumulate dirt and chips, and is therefore not equipped, as a rule, with shields or other protecting devices. The encompassed type retains lubricant poorly, in contrast to the encompassing type (with the apex downwards).

Vee ways are made symmetrical if, for example, the load is directed vertically, as from the weight of the travelling unit. They may be unsymmetrical with one larger face which, in this case, is located perpendicular to the direction of the resultant external load. Most lathes have this type of ways.

Dovetail ways, or *guides* (Fig. 129*c*), are distinguished for the small space they occupy and their comparatively simple clearance adjustment by means of a single taper or flat gib (see, for instance, Fig. 132).

Cylindrical (*bar-type*) *ways*, or *guides* (Fig. 129*d*), notwithstanding their simple manufacture, are relatively seldom used in machine tools. Their main shortcoming is the low rigidity which results from the fact that they are secured to the bed only at their ends. Besides, quite complex devices are required to adjust clearances in cylindrical ways.

Combination ways, commonly used in machine tools, have one flat way while the other is prismatic, being either a vee way or shaped like one half of a dovetail. Such combination ways are comparatively producible and especially suitable in cases when the unit is subject to large overturning moments.

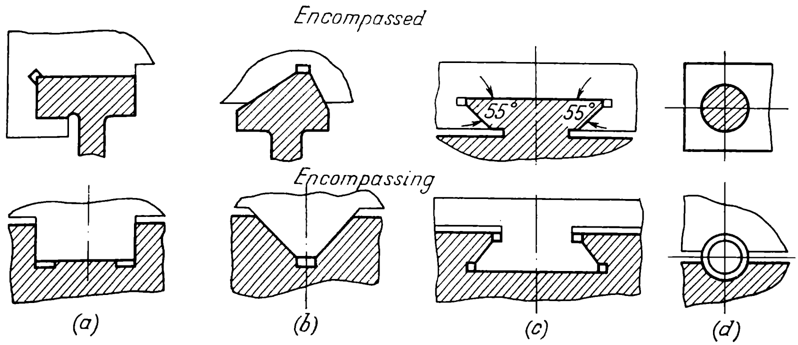


Fig. 129. Principal shapes of machine tool slideways:
(*a*) flat; (*b*) vee; (*c*) dovetail; (*d*) cylindrical (bar-type)

The final choice of the type of ways to employ in designing a new machine tool should be based on the possibility of ensuring their maximum rigidity under the action of loads which are representative of the given type of machine tool.

Devices for Adjusting Clearances in Slideways

Optimum clearances in slideways, ensuring accuracy of travel with minimum friction losses, are difficult to maintain in manufacture even if the mating surfaces are fitted to each other. Moreover, the initially adjusted clearances are altered in the course of wear of the sliding surfaces. For this reason, ways and guides are equipped with devices for periodically adjusting the clearances between the mating surfaces.

The most general solution of the problem of adjusting clearances in ways is illustrated in Fig. 130. The clearances between the contacting horizontal surfaces, carrying the vertical pressure V, are adjusted by flat gibs *1* and *2*. The clearances between the vertical contacting surfaces, carrying the horizontal pressure H_1 or H_2 and constituting the guiding surfaces proper, are adjusted by taper gib *3*.

If the saddle or slide encompasses the contour of flat ways of the bed only on three sides (Fig. 131), flat gibs *1* and *2*, secured to the slide by screws, are required. Scraping will be required to compensate for wear of the horizontal faces. Sometimes, to avoid scraping, thin shims are used (see part *1* in Fig. 134*b*). Flat gib *4*, of constant thickness, is used here to adjust the clearance in the vertical contacting surfaces. In the course of its wear, this

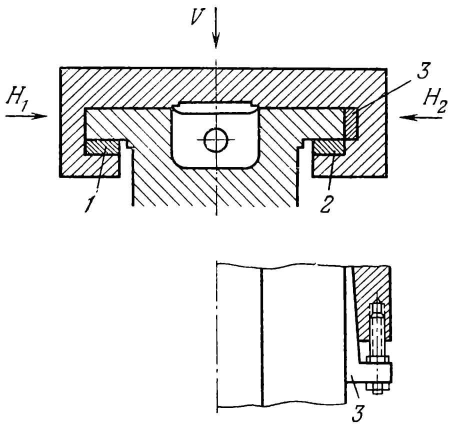


Fig. 130. Adjusting clearances in ways (general case)

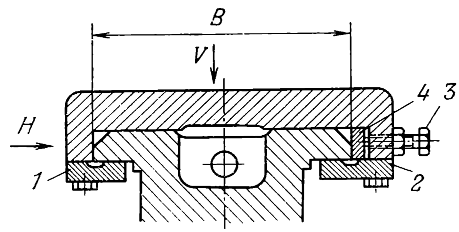


Fig. 131. Adjusting clearances in flat ways

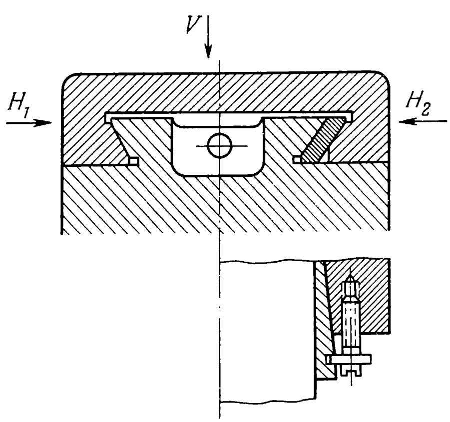


Fig. 132. Adjusting clearances in dovetail ways or guides

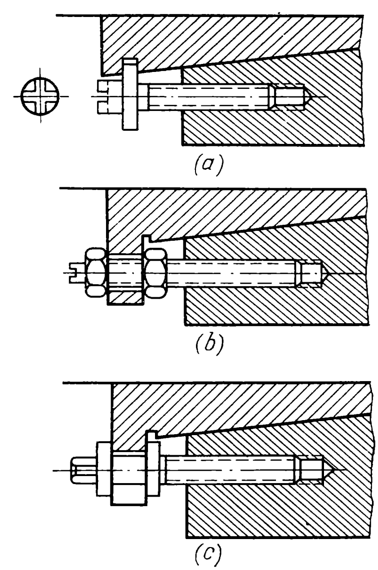


Fig. 133. Regulating screws for taper gibs

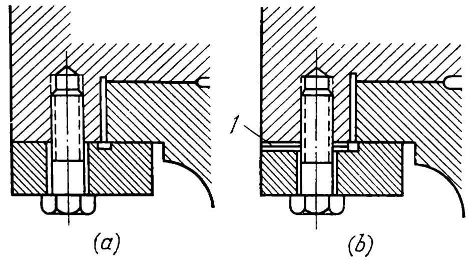


Fig. 134. Methods of adjusting flat gibs

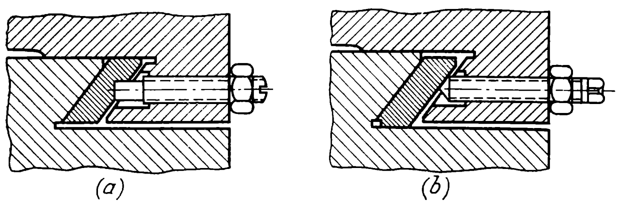


Fig. 135. Parallel gibs for clearance adjustment in dovetail ways

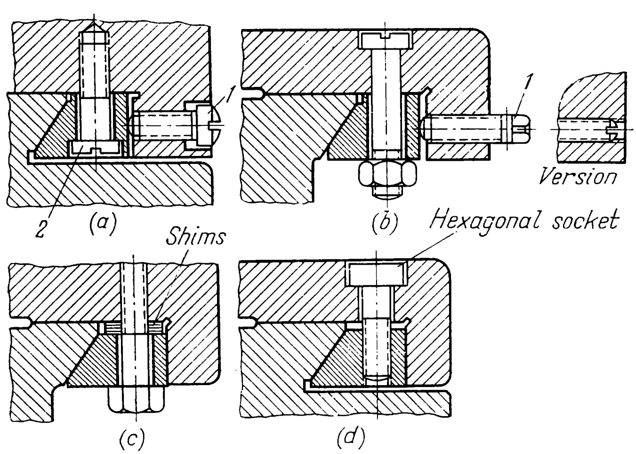


Fig. 136. Gibs of trapezoidal cross section for dovetail ways

gib is adjusted forward by several screws *3*. A taper gib could be used here as well in place of the flat gib.

The method of adjustment used for dovetail ways or guides is shown in Fig. 132.

In clearance adjustment with a flat gib of constant thickness, the gib should be arranged so that its pressure is carried by the directly contacting faces of the ways. This means that the gib should be on the side or way opposite to the one to which the load is applied (as in the case of load H in Fig. 131). If a taper gib is employed, it may be on either side (as in the case of loads H_1 and H_2 in Fig. 130).

Taper gibs usually have an inclination in the range from 1 : 40 to 1 : 100. The longer the gib, the less it is tapered. To adjust the gib, facilities should be provided for moving it in both directions. Various designs of regulating screws can be used for this purpose; the most common designs are shown in Fig. 133*a*, *b* and *c*.

The various shapes of flat and taper gibs employed in up-to-date machine tools are shown in cross section in Figs. 134, 135 and 136. They need no further explanation.

The general shortcomings of adjustable taper gibs are that they increase the number of joints between the mating parts, that they themselves possess low rigidity and, as a result, lower the rigidity of the unit in respect to compressive forces. The effect of these shortcomings can be reduced to some extent by correctly locating the gibs and by providing means for clamping them tightly after making adjustments.

Attached Ways

Attached ways are usually of steel but in some cases they are made of high-quality cast iron. The ways are designed as strips secured to a cast iron bed with screws or welded to a welded steel bed.

When ways are secured mechanically, the design of the fastening should be such that no damage is done to the working surface of the ways. This is shown in Fig. 137*a*. If considerations of design exclude fastening from underneath with screws, a fastener should be used which does not violate the homogeneity of the working surfaces. For example, the screws shown in Fig. 137*b* are made of the same material as the attached ways. After tightly screwing in the screws, the heads are cut or broken off at the narrow neck and the remaining part of the screws are ground off flush with the way surface.

Ways secured with screws usually have an integral key (Fig. 137*c*) which relieves the screws of lateral loads and considerably increases the transverse rigidity of the ways.

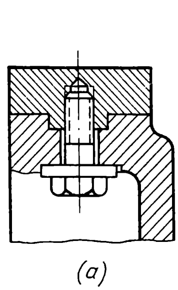


Fig. 137. Attached ways

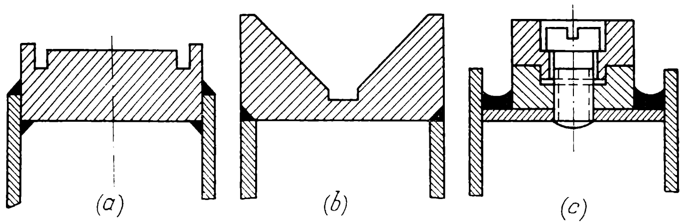


Fig. 138. Welded ways

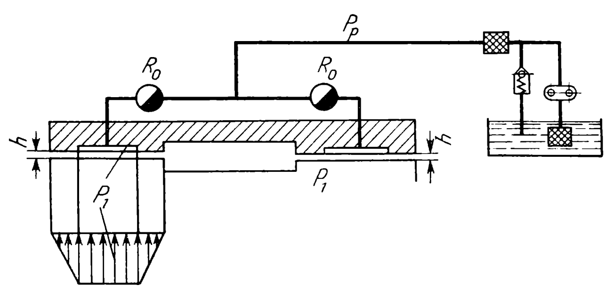


Fig. 139. Principle of hydrostatically lubricated slideways

Examples of ways welded to a welded steel bed are shown in Fig. 138. In the last case (Fig. 138*c*), a cast iron or bronze attached way is to be secured to the welded steel bed.

Ways of plastics are usually secured by screws but are sometimes glued to the bed.

10-2. Hydrostatically Lubricated Slideways

Slideways with provision for delivering oil under pressure between the mating surfaces, so as to produce an oil film over the full contact area, are called *hydrostatically lubricated slideways.*

From the pump (Fig. 139) oil is delivered under pressure through flow-control valves with a restriction R_0 into pockets made in the ways. From the pockets the oil escapes through the clearance h between the slideway surfaces. In this clearance the oil pressure varies according to an approximately linear function.

The load-carrying capacity of hydrostatically lubricated slideways can be calculated by the equation

$$P = p_1 F \alpha \tag{196}$$

where p_1 = oil pressure in the pockets
F = area of the slideways
α = factor taking into consideration the drop in oil pressure in the clearance, and approximately equal to

$$\alpha = \left(\frac{1}{3} + \frac{l}{6L} + \frac{b}{6B} + \frac{lb}{3LB}\right) = \frac{1}{3} \text{ to } \frac{1}{2} \tag{197}$$

Figure 140 illustrates the commonly used types of oil pockets and the points to which oil under pressure is delivered. The first version, with a single longitudinal groove, is employed for narrow slideways, while versions II and III are suitable for wider slideways (over 50 or 60 mm). The main design parameters can be determined from the following relationships

$$a_1 \cong 0.1B \quad a \cong 0.5a_1 \quad a_2 \cong 2a_1$$

Several pockets with independent oil delivery should be provided along the length of the way as otherwise high rigidity cannot be ensured under the action of skew moments.

The rigidity of hydrostatically lubricated slideways is directly proportional to the normal force and inversely proportional to the magnitude of the clearance. Thus

$$j = 3\left(1 - \frac{p_1}{p_p}\right)\frac{P}{h} \tag{198}$$

where p_p is the pressure of the oil delivered by the pump (see Fig. 139).

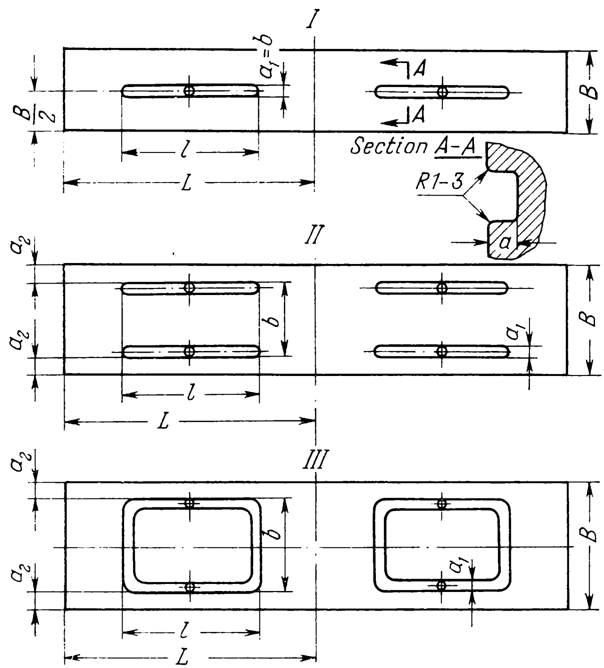


Fig. 140. Methods of oil input in hydrostatically lubricated slideways

Consequently, to attain a high rigidity in hydrostatically lubricated slideways, it is necessary to make the clearance h as small as possible. This clearance depends upon the macro- and microirregularities of the slideway surfaces. With high-quality scraping [16 to 20 spots in an area (25×25) sq mm] a minimum design clearance of 15 to 25 microns can be maintained. This provides a rigidity of hydrostatically lubricated slideways in the order of 100 kgf per micron and even more.

Any shape of slideway can be hydrostatically lubricated. Thus, for instance, a combination of one vee and one flat way is often used for this purpose in grinding machines (Fig. 141).

Air lubricated slideways have also been used to some extent in machine tools. Here an air cushion is produced in the clearance between the mating bearing surfaces. Air from the compressed air mains passes through a filter and pressure regulating valve and enters the pockets at a pressure of 3 or 4 kgf per sq cm through apertures of small diameter (0.2 to 0.5 mm) as shown in Fig. 142.

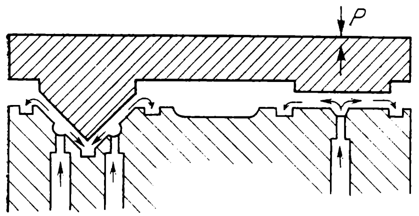


Fig. 141. Combination (flat and vee) hydrostatically lubricated slideways

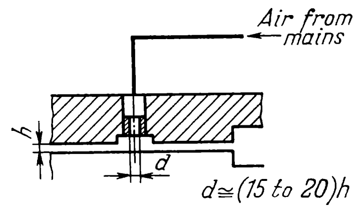


Fig. 142. Air lubricated slideways

10-3. Slideway Design

The wear resistance of slideways depends upon various conditions, one of the most important being as uniform as possible distribution of the pressure over the way surfaces, the average (conditional) specific pressure not exceeding a certain definite value established on the basis of experience in machine tool operation (see p. 224). The specific pressure is determined by checking calculations based on the assumption that the specific pressure is distributed according to a linear function lengthwise along the slideway; across the width of each face of the slideway, the specific pressure is considered to be distributed uniformly.

A scientifically grounded procedure for designing slideways, one sufficiently reliable in practice, was first developed by D. Reshetov (ENIMS) in the USSR in 1942. The essentials of this procedure, accepted at the present time as a machine tool industry standard (Std H49-2MCC), are set forth below for the case of combination ways of a lathe. With certain revisions this procedure is applicable to ways of other shapes.

This procedure consists of the following stages:

(1) determining the total pressure acting on each face of the ways;

(2) determining the average specific pressure on each face of the ways;

(3) determining the maximum specific pressure on the faces;

(4) comparing the calculated values with maximum permissible specific pressures, known from experiments.

The pressures on the faces of the bed ways, or the three reactions A, B, and C (Fig. 143), equal in magnitude to these pressures, can be found from the conditions of equilibrium of the carriage. Acting on the carriage, in addition to these reactions, are: (a) the components P_z, P_x and P_y of the cutting force; (b) dead weight G of the carriage, considered as a force concentrated at the centre of gravity; (c) traversing force Q, and the frictional forces fA, fB and fC, acting on the faces of the ways in a direction opposed to carriage travel.

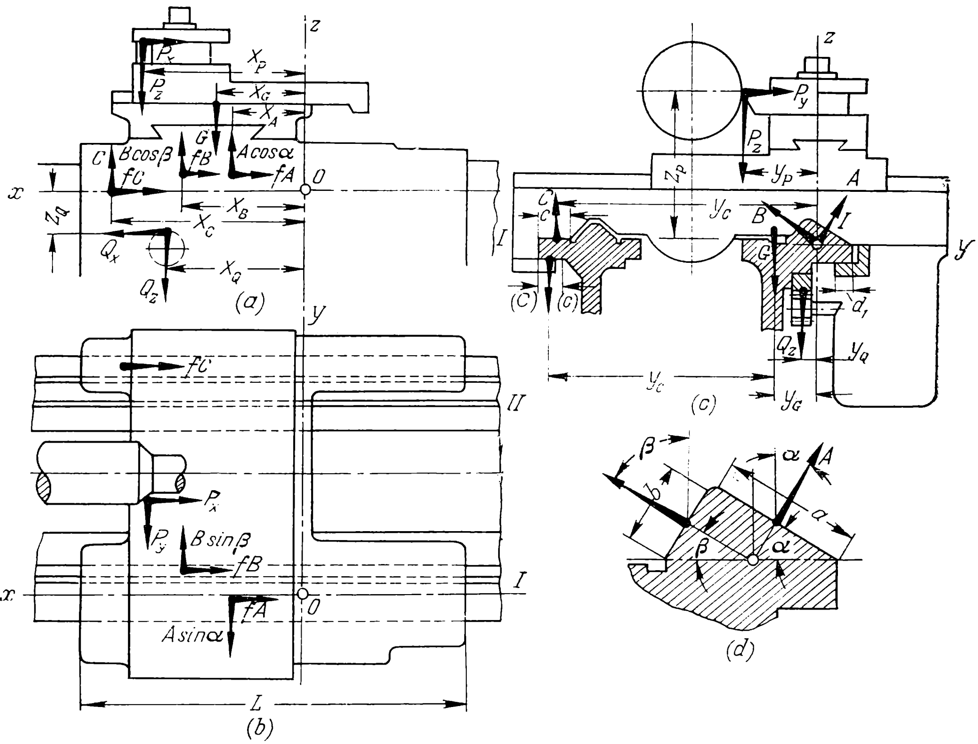


Fig 143. Design diagram for the slideways of a lathe

The components forces P_z, P_x and P_y are either calculated by formulas of metal-cutting theory or they are taken according to reference data for a speed, feed and depth of cut that completely utilize the power capacity of the machine tool. The weight G of the carriage and its centre of gravity are found by calculations, and if possible on a model. If the weight of the workpiece and of the fixture are also carried by the ways, these forces should be taken into consideration.

If the carriage is traversed by a lead screw, the pulling force Q, required to traverse the carriage, is directed along the axis of the lead screw. Therefore, this force has no components parallel to the forces P_y and P_z. If, on

the other hand, the carriage is traversed from a feed rod through a spur pinion and rack, there will be, in addition to the component Q_x parallel to the feed force P_x, another component

$$Q_z = Q_x \tan (\alpha + \rho)$$

where α = pressure angle of the rack pinion
ρ = 5° to 7° = angle of friction on the teeth.

The axes of the co-ordinates x, y and z in Fig. 143 are selected parallel to the components P_x, P_y and P_z of the cutting force, respectively, while the origin O of the co-ordinates is at the point of intersection of reactions A and B. This has been done to keep the equations of equilibrium of the carriage as simple as possible. Using the arrangement shown in Fig. 143 for the case being considered we can readily write six equations of equilibrium

$$\left.\begin{array}{lll} \sum X = 0 & \sum Y = 0 & \sum Z = 0 \\ \sum M_x = 0 & \sum M_y = 0 & \sum M_z = 0 \end{array}\right\} \qquad (199)$$

The forces enumerated above are substituted in the left-hand side of these equations; these forces include the unknowns A, B, C, and Q. These can be determined using the first four equations of the system (199).

When the forces A, B and C have been found, there will be no trouble in determining the average specific pressures. Thus

$$p_{A_{av}} = \frac{A}{aL}; \quad p_{B_{av}} = \frac{B}{bL} \text{ and } p_{C_{av}} = \frac{C}{cL} \qquad (200)$$

where L = length of the carriage ways
a, b and c = working widths of the three faces of the slideways (see Fig. 143).

To determine the maximum specific pressures it will be necessary to find the three co-ordinates x_A, x_B and x_C of the points of application of the resultant forces A, B and C. Only the last two equations of the system (199) have not yet been used. They can be written (see Fig. 143) as

$$\left.\begin{array}{l} Ax_A \cos\alpha + Bx_B \cos\beta + Cx_C = \overline{M}_y \\ -Ax_A \sin\alpha + Bx_B \sin\beta = \overline{M}_z \end{array}\right\} \qquad (201)$$

where, for the sake of brevity, the following notation has been introduced (see Fig. 143)

$$\left.\begin{array}{l} \overline{M}_y = -P_x z_p + P_z x_p + G x_G - Q_x z_Q + Q_z x_Q - f(A+B+C)\,s \\ \overline{M}_z = -P_x y_p + P_y x_p + Q_x y_Q - f(A+B+C)\,t \end{array}\right\} \qquad (202)$$

To find the co-ordinates x_A, x_B and x_C from equations (201), it is necessary to establish the distribution of moment $\overline{M}_y$ between the front (*I* in Fig. 143*b*) and rear (*II*) ways. This distribution depends upon the rigidity of the carriage, the degree of nonuniformity of the load on the bed ways (triangular, trapezoidal or other load) and the shape of the ways. For example, in the case of flat ways, the moments M_{I} and M_{II} are proportional to the widths of the ways; in case of combination ways, one moment is proportional to the width of the flat way while the other is proportional to the equivalent width of the vee way, etc. This question is treated in more detail in the book *The Design of Machine Tool Components* (ENIMS, 1945) by D. Reshetov who developed this design method.

Let us assume that the distribution of moment $\overline{M}_y$ between the front and rear ways has been established, i.e., the corresponding moments M_{I} and $M_{\mathrm{II}} = \overline{M}_y - M_{\mathrm{I}}$ have been determined. Then the first of equations (201) can be broken down into two equations, so that a system of linear equations is obtained

$$\left.\begin{aligned} &Ax_A \cos\alpha + Bx_B \cos\beta = M_{\mathrm{I}} \\ &Cx_C = M_{\mathrm{II}} \\ &-Ax_A \sin\alpha + Bx_B \sin\beta = \overline{M}_z \end{aligned}\right\} \tag{203}$$

hence

$$\left.\begin{aligned} x_A &= \frac{M_{\mathrm{I}} \sin\beta - \overline{M}_z \cos\beta}{A \sin(\alpha+\beta)} \\ x_B &= \frac{M_{\mathrm{I}} \sin\alpha + \overline{M}_z \cos\alpha}{B \sin(\alpha+\beta)} \\ x_C &= \frac{M_{\mathrm{II}}}{C} = \frac{\overline{M}_y - M_{\mathrm{I}}}{C} \end{aligned}\right\} \tag{204}$$

In case of linear distribution of the specific pressure along the face of the ways, the ratios $\frac{x_A}{L}$, $\frac{x_B}{L}$ and $\frac{x_C}{L}$ determine the shape of the pressure diagram. For the most general case—trapezoidal distribution of the pressure (Fig. 144*a*)—the distance of the point of application of the resultant A of the pressure p_A from the larger base of the trapezoid is

$$\frac{L}{2} - x_A = \frac{L}{3}\,\frac{p_{A\,max} + 2p_{A\,min}}{p_{A\,max} + p_{A\,min}}$$

then

$$x_A = \frac{L}{6}\,\frac{p_{A\,max} - p_{A\,min}}{p_{A\,max} + p_{A\,min}} \tag{205}$$

Hence, it follows that if $0 < x_A < \frac{L}{6}$, then the diagram of specific pressure p_A has the form of a trapezoid.

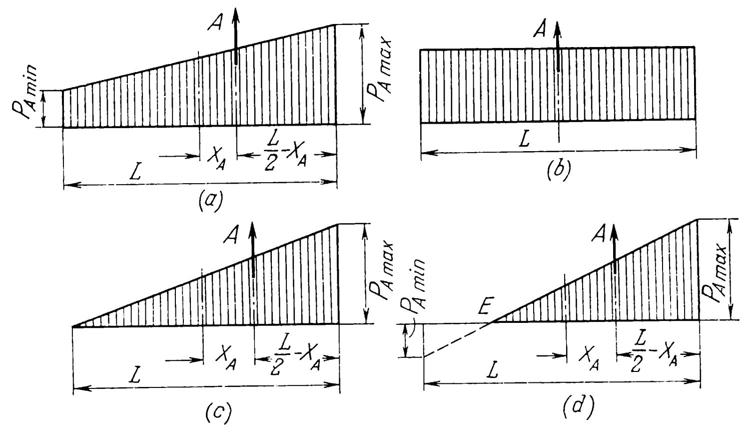


Fig. 144. Diagrams of pressure distribution along the length of the ways

Similar equations can be obtained for the co-ordinates x_B and x_C of the centres of pressure p_B and p_C, therefore the subindex A is omitted in the following.

At $x = 0$, equation (205) becomes

$$p_{max} = p_{min}$$

which means that the specific pressure p is distributed along the way according to a rectangle (Fig. 144*b*), i.e.,

$$p = \text{const}$$

If $x = \frac{L}{6}$, it follows from the same equation that

$$p_{max} - p_{min} = p_{max} + p_{min}$$

hence

$$p_{min} = 0$$

and the pressure distribution is according to a triangle (Fig. 144*c*).

Finally, if it turns out after solving equations (204) that $x > \frac{L}{6}$, in which case, formally $p_{min} < 0$, this means that the bed and saddle ways are in contact only over a part of the length L, as shown in Fig. 144*d*. To the left of point E the joint between the ways is relieved of a load because of the large clearance between the gib and the lower face of the ways.

Having calculated the average specific pressures p_{av} from equations (200) and the values of the co-ordinates x from equations (204) and knowing, there-

fore, the shape of the diagram of specific pressures p from the ratio $\frac{x}{L}$, the maximum specific pressure p_{max} can be determined for each face of the ways. In case of trapezoidal distribution of p (Fig. 144a), from the equations $p_{max} + p_{min} = 2p_{av}$ and $p_{max} - p_{min} = \frac{6x}{L} 2p_{av}$, we obtain

$$p_{max} = p_{av}\left(1 + \frac{6x}{L}\right) \tag{206}$$

In case of triangular distribution of p over a part of the length, it follows from the diagram in Fig. 144d that

$$p_{av}L = \frac{1}{2} p_{max} 3\left(\frac{L}{2} - x\right)$$

hence

$$\left.\begin{aligned} p_{max} &= p_{av}\frac{2L}{1.5L - 3x} \\ \text{or} \quad p_{max} &= \frac{4}{3} p_{av} \frac{1}{1 - 2\frac{x}{L}} \end{aligned}\right\} \tag{207}$$

For triangular distribution along the full length L (Fig. 144c), which can be regarded as the limiting case of both of the preceding types of distribution, after substituting the value $\frac{x}{L} = \frac{1}{6}$ in equations (206) and (207) we can write

$$p_{max} = 2p_{av} \tag{208}$$

If in equations (206), (207) and (208) we substitute the value of p_{av} from equation (200), we obtain for face A

$$\left.\begin{aligned} p_{A\,max} &= \frac{A}{aL}\left(1 + \frac{6x}{L}\right) \text{ for } x \leqslant \frac{L}{6} \\ p_{A\,max} &= \frac{A}{aL}\,\frac{2L}{1.5L - 3x} \text{ for } x \geqslant \frac{L}{6} \end{aligned}\right\} \tag{209}$$

Similar equations are obtained for $p_{B\,max}$ and $p_{C\,max}$.

A procedure similar to or resembling that set forth here is employed for the checking calculations of slideways of other shapes.

Machine tool industry standard H49-2 lists the following permissible p_{max} values for cast iron slideways:

(a) at low sliding speeds in the order of the rates of feed (lathes and milling machines), p_{max} = 25 to 30 kgf per sq cm;

(b) at high sliding speeds, in the order of the cutting speeds (planers, shapers and slotters), p_{max} = 8 kgf per sq cm;

(c) for special-purpose machine tools, operating at constant heavy feeds and high speeds, the specified values of p_{max} should be reduced by approximately 25 per cent;

(d) for heavy machine tools, $p_{max} = 10$ kgf per sq cm for low sliding speeds and $p_{max} = 4$ kgf per sq cm for high sliding speeds.

A value $p_{max} = 0.5$ to 0.8 kgf per sq cm is suitable for the slideways of grinding machines.

If checking calculations are limited to a determination of only average values of the specific pressure, it is recommended that the permissible average values be taken one half of the above-listed p_{max} values.

Because of the comparatively short time they have been employed, it has not been possible to establish permissible p_{max} values for steel slideways. In a combination of steel on cast iron ways, the p_{max} values are about the same as for cast iron on cast iron. In the case of steel ways on steel ways, these permissible values can be increased by 20 to 30 per cent.

10-4. Antifriction Ways

The main advantage of antifriction ways, as their name implies, is the low friction which does not practically depend upon the speed of travel. This ensures highly sensitive precision movements and uniform slow motion. In addition, antifriction ways have a considerably longer service life than slideways.

Drawbacks of antifriction ways include their higher cost, necessity for more accurate machining of the working surfaces and, finally, the lagging behind of the rolling elements from the traversed unit (Fig. 145). Therefore, in designing antifriction ways for long distances of travel, it is necessary to provide facilities for return or recirculation of the rolling members. Antifriction ways can be of open and closed design as in the case of slideways.

The friction force in open antifriction ways can be expressed by the equation

$$T = nT_0 + \frac{f_r}{r_{eq}} P \qquad (210)$$

where T_0 = constant component on one face of the ways not dependent upon the normal force

n = number of faces (races) constituting the ways

f_r = coefficient of rolling friction, equalling approximately 0.001 cm for ground steel ways and approximately 0.0025 cm for scraped cast iron ways

r_{eq} = equivalent radius of the rolling members, cm

P = normal force.

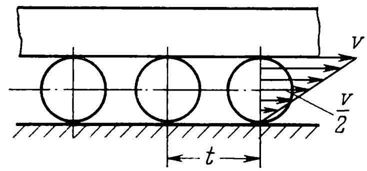


Fig. 145. Rolling member velocity in antifriction ways

The preload should be also taken into consideration in case of closed antifriction ways.

Table 3 illustrates the most widely employed types of antifriction ways and the formulas for calculating the required traversing force. The first three are open, and the last three closed types of ways. The friction force in antifriction ways does not usually exceed 1 to 4 kgf.

Constructions of Antifriction Ways

Open antifriction ways using balls (Fig. 146*a*) or rollers (Fig. 146*b*) find application in cases when the main load constitutes the dead weight of the travelling unit and varies only slightly during the machining operation.

Closed antifriction ways (Fig. 147*a*, *b* and *c*) incorporate means for preloading and have a much higher rigidity. Preloading is accomplished by taper gibs or adjustable flat gibs, in much the same manner as the clearance is adjusted in slideways.

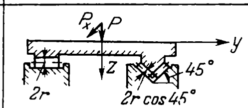
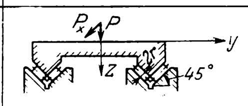
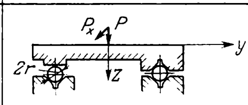
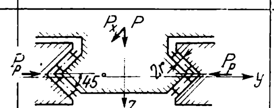
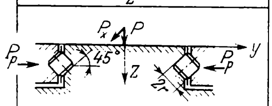
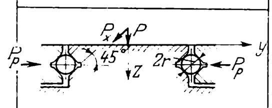
To eliminate the principal drawback of antifriction ways, the lagging behind of the rolling members, recirculation of the balls or rollers is used in various constructions. In the example shown in Fig. 148*a* the balls are arranged in a continuous row between four cylindrical rods, of which two are secured to the stationary bed and two to the travelling unit. Guides *1*, along which the balls enter the return channel *2*, are provided at the ends of the ways. In the second example (Fig. 148*b*), blocks with rollers are used. These blocks are arranged at the end of the ways and have facilities for recirculation of the rollers in reference to the block.

Antifriction Way Design

The design of antifriction ways consists of strength calculations based on the contact stresses; the contact rigidity is additionally calculated in designing precision machine tools.

TABLE 3

Traversing Force Calculations

	Type of ways	r_{eq}, cm	Traversing force Q, kgf
1		$\frac{r}{1.5}$	$Q=P_x+3T_0+\frac{1.5}{r}f_r P$ $P=P_z+G_1+G_2$
2		$\frac{r}{1.4}$	$Q=P_x+4T_0+\frac{1.4}{r}f_r P$
3		$\frac{r}{1.5}$	$Q=P_x+2T_0+\frac{1.5}{r}f_r P$
4		$\frac{r}{2.8}$	$Q=P_x+4T_0+\frac{2.8}{r}f_r P_p$
			$Q=P_x+2T_0+\frac{2.8}{r}f_r P_p$
			

Notes: 1. The coefficient of rolling friction $f_r = 0.001$ for ground steel ways and $f_r = 0.0025$ for scraped cast iron ways. The initial friction force, referred to one separator, $T_0 = 0.4$ kgf.

2. Because of the low value of the friction forces, a simplified arrangement has been accepted in which the ways are subject only to the feed force P_x, vertical component P_z of the cutting force, table weight G_1 and workpiece weight G_2. The tilting moments, force P_y and the components of the traversing force are not taken into account.

3. In the type 4 ways only the feed force P_x and the preload force P_p are taken into consideration.

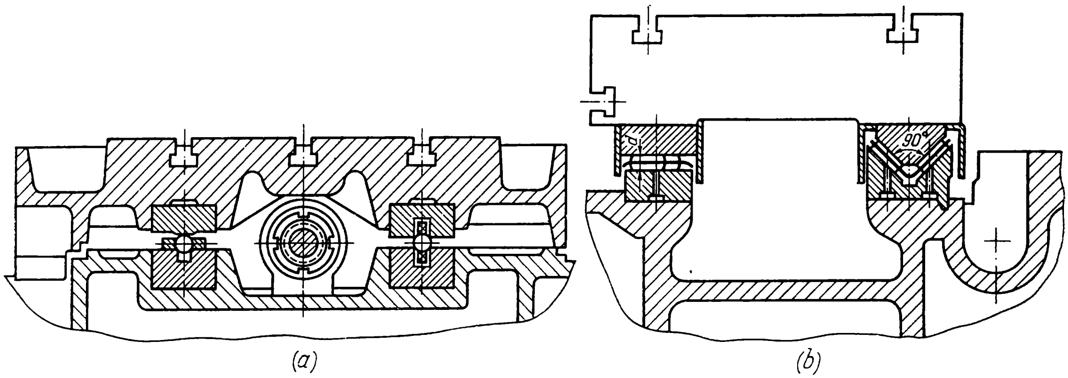


Fig. 146. Open antifriction ways:
(*a*) ball; (*b*) roller

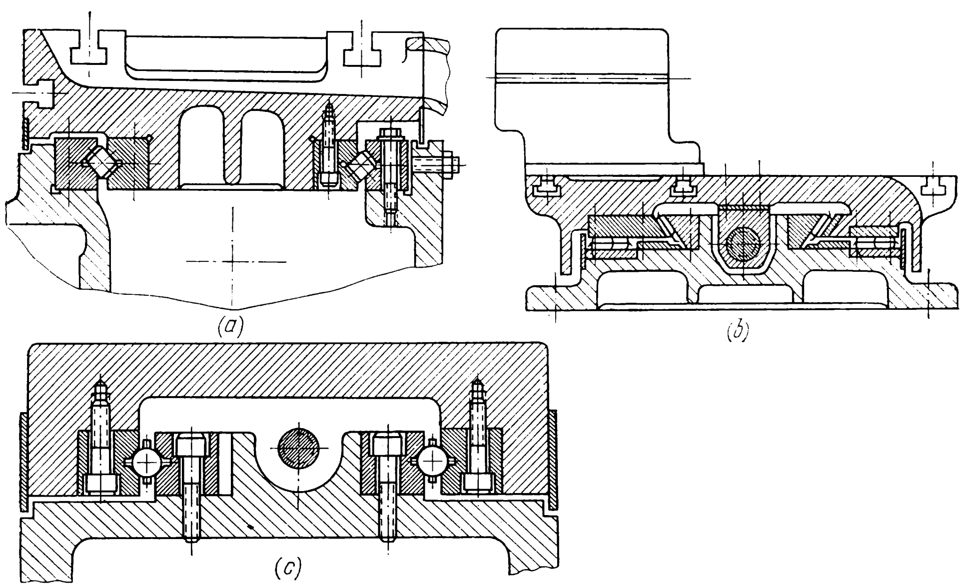


Fig. 147. Closed antifriction ways:
(*a*) roller vee; (*b*) roller dovetail; (*c*) ball vee

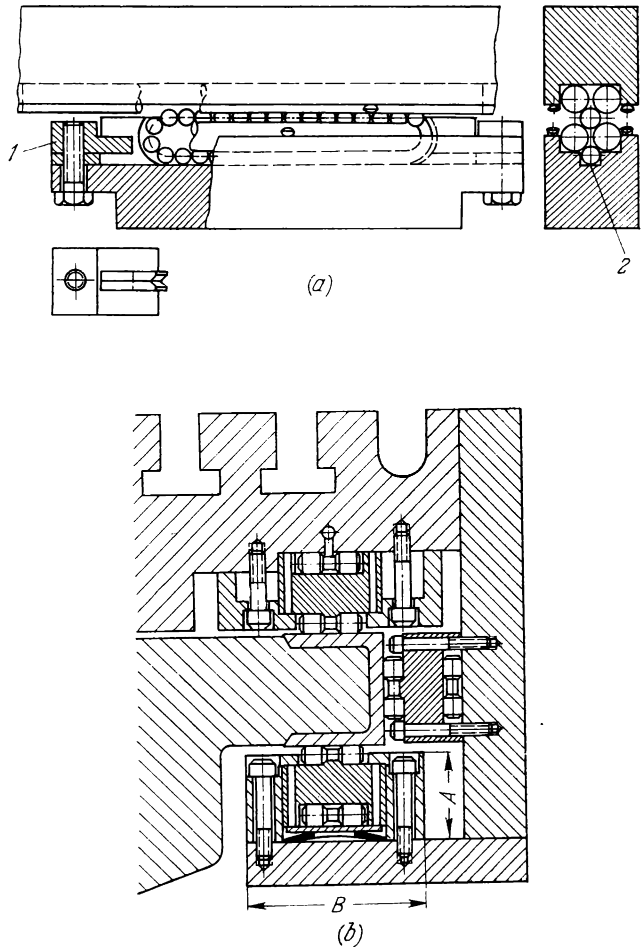


Fig. 148. Antifriction ways with recirculation of the rolling members

In strength calculations it is first necessary to determine the load acting on the most heavily loaded ball or roller. Investigations of D. Reshetov, E. Rivin and Z. Levina show that the formulas for calculating the pressure in slideways can be used for this purpose. Thus, the maximum compressive force acting on the rolling member is

$$p_{max} = p_{max}tb \tag{211}$$

where p_{max} = maximum pressure in slideways
t = pitch of the rolling members (see Fig. 145)
b = width of the ways.

The permissible load for roller ways, based on the conditions of contact strength, is

$$P_{per} = \sigma_c \, db \tag{212}$$

while for ball ways it is

$$P_{per} = \sigma_c d^2 \tag{213}$$

where σ_c = conditional stress, referred to the sectional area of the rolling member
d = diameter of the ball or roller
b = length of the roller.

The permissible conditional stress σ_c for ball ways is 6 kgf per sq cm for steel ways (ball races) hardened to $60R_C$, and 0.2 kgf per sq cm for cast iron ways (Bhn 200). The permissible stress for roller ways ranges from 150 to 200 kgf per sq cm for hardened steel ways (races) and from 13 to 20 kgf per sq cm for cast iron ways.

The above calculations for checking the contact strength of antifriction ways does not take into consideration the effect of errors in making the ways (straightness errors) nor the differences in size of the rolling members. Therefore, if the manufacturing accuracy of the ways is not very high, when the total deviation from straightness over the length of contact is in the order of 15 to 20 microns, the value of σ_c should be reduced by 20 to 30 per cent.

Rigidity calculations for antifriction ways consist in determining the elastic displacements due to contact deformation under the action of the external load. In this case it is of especial importance to take into consideration the effect of the manufacturing errors on the character of the load distribution among the rolling members, since these errors are of the same order as the elastic deformations. Fig. 149 shows the effect of errors in antifriction ways upon the distribution of the pressure among the rolling members.

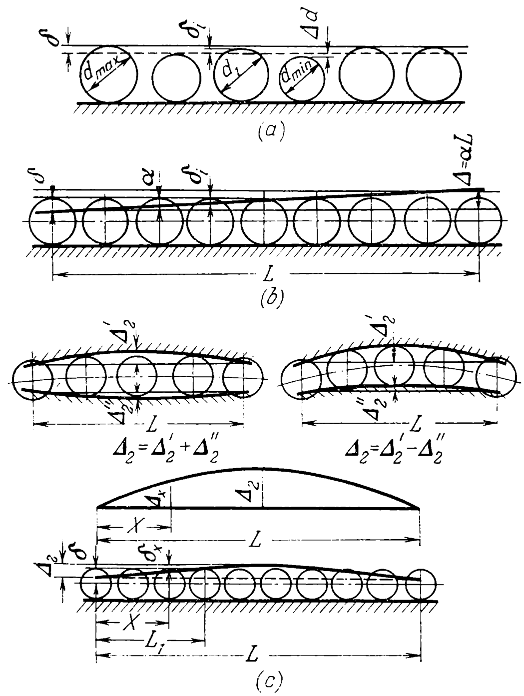


Fig. 149. Effect of errors in the ways on the load distribution among the rolling members: (a) difference in size of the rolling members; (b) lack of parallelism of the ways; (c) lack of straightness in the ways

In engineering calculations for determining the rigidity, the elastic displacements in antifriction ways can be found by the following equations:

for roller ways

$$\delta = c_r q \tag{214a}$$

and for ball ways

$$\delta = c_b P \tag{214b}$$

where δ = elastic displacement, microns
c_r = coefficient of unit deflection of roller ways, micron-cm per kgf
c_b = coefficient of unit deflection of ball ways, microns per kgf
q = running (linear) load per unit of roller length, kgf per cm
P = load on one ball, kgf.

The values of the coefficients of unit deflection for antifriction ways, manufactured to standard accuracy, are given in Fig. 150.

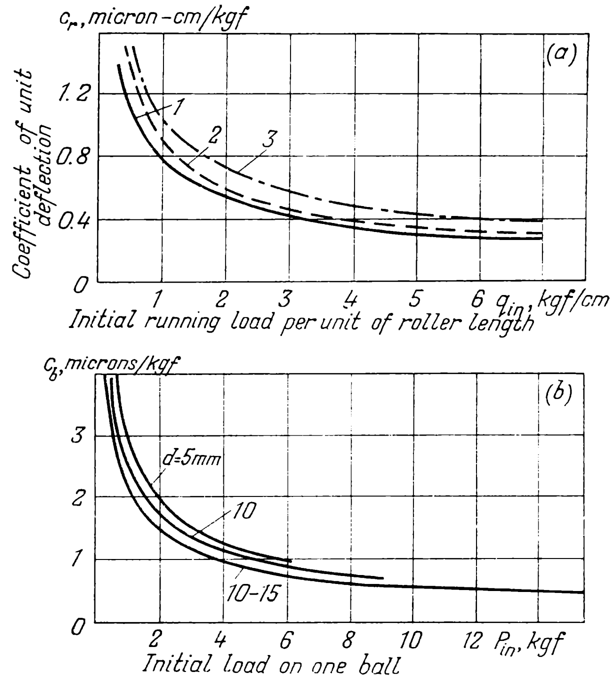


Fig. 150. Coefficients of unit deflection of antifriction ways:
(a) roller ways; (b) ball ways; 1 and 2—ground steel ways with short and long rollers, respectively; 3—scraped cast iron ways

Calculations and experimental data show that the rigidity of roller ways approaches that of slideways and may even be three or four times higher if they are suitably preloaded. The rigidity of ball ways is only from 40 to 50 per cent that of roller ways if the balls and rollers are of the same diameter.

The surface roughness of the ways (races) has a marked influence on the rigidity of antifriction ways. Therefore, strict requirements are specified for the scraping of such ways. It proves expedient to resort to lapping in manufacturing critical antifriction ways for precision machine tools.

10-5. Circular Ways

Circular ways are employed in various machine tools for the main cutting motion (in vertical turret lathes and vertical turning and boring mills), as well as for the work speed motion (in hobbing machines and surface grinders) and sometimes for handling motions.

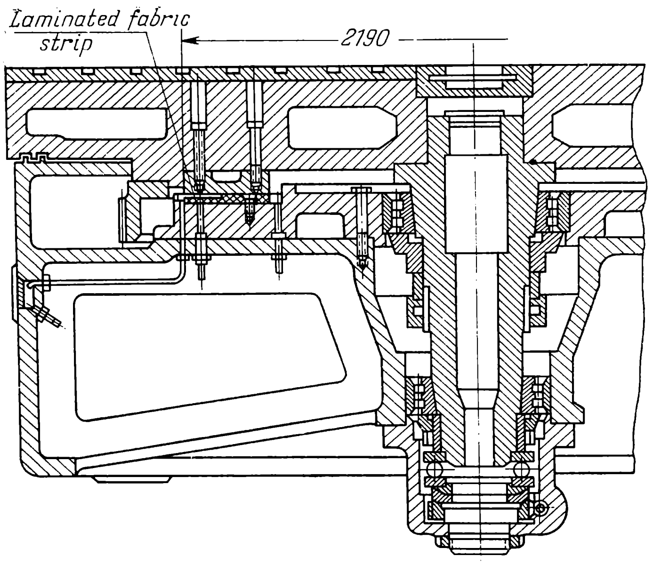


Fig. 151. Flat circular ways of a vertical boring mill with laminated fabric strip

The general principles that underlie the selection of the type of ways and their construction do not differ essentially from those set forth above for straight-line bearings or ways.

Flat ways (Fig. 151) are most frequently used because of manufacturing considerations, since they are simpler to machine and to assemble. Their application is especially justified in cases when a central bearing locates the axis of rotation of the table and the circular ways carry only the vertical load.

If there is no central bearing, a vee-type circular way, usually of unsymmetrical profile (Fig. 152) is employed.

If the rotary table is of very large size, as in heavy vertical turning and boring mills, two circular ways are used. This is done to reduce the vertical deformation of the table by providing intermediate bearing surfaces in the ways.

Circular antifriction ways (Fig. 153) are employed in high-speed vertical turret lathes. They possess the same features as antifriction ways for straight-line motion.

The circular motion limits the application of rollers in antifriction ways, consequently, ball-type circular ways are more widely used in machine tools.

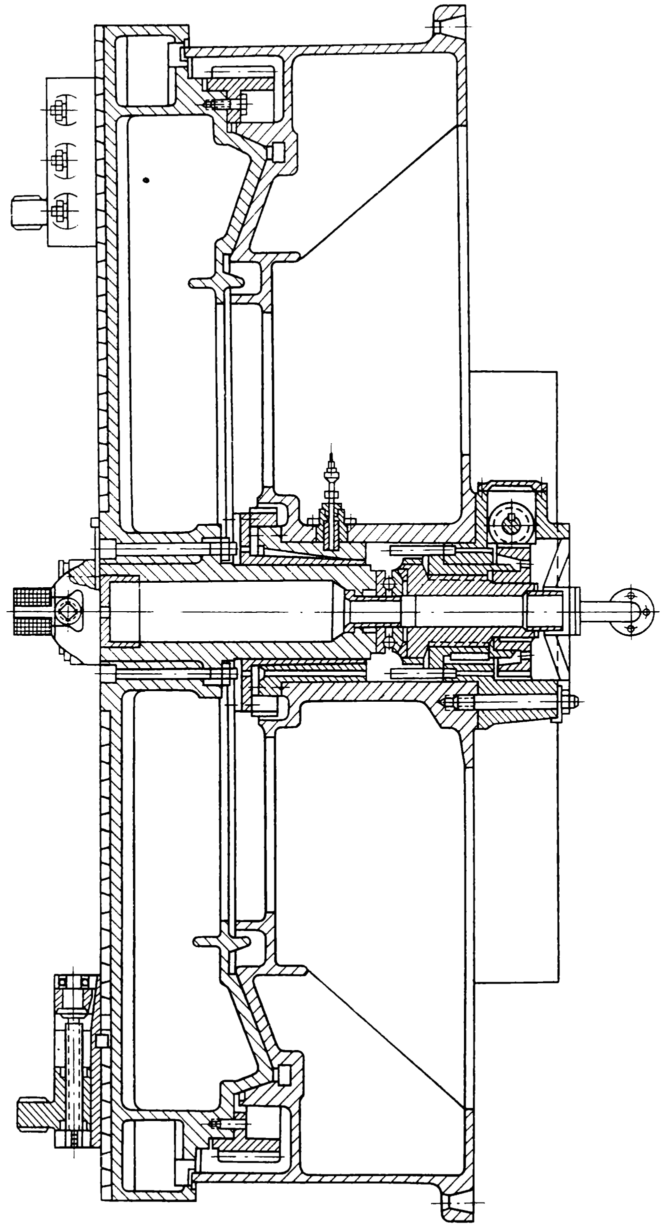


Fig. 152. Vee-type circular ways of a vertical turning and boring mill

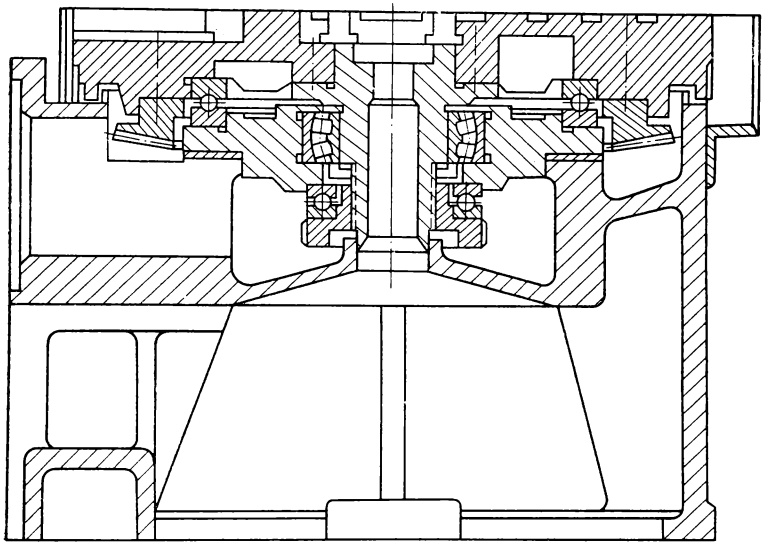


Fig. 153. Circular antifriction ways

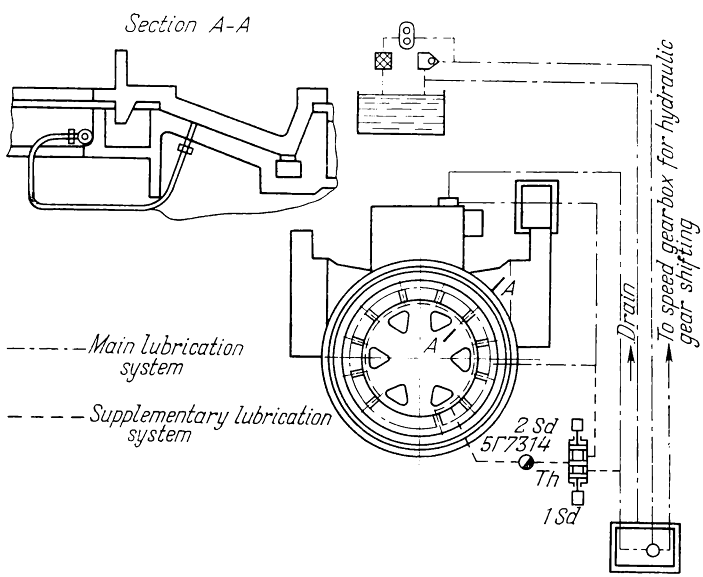


Fig. 154. Circular ways with hydrostatic load relief

Circular ways are partially relieved of their load in order to reduce the contact pressure and consequent wear by the provision of an additional adjustable antifriction thrust bearing and the delivery of lubricant under pressure between the working surfaces of the ways. A system of hydrostatically lubricated circular ways can be set up if lubricant of sufficiently high surplus pressure is available and several independent pockets are provided.

Figure 154 shows the lubrication system of the model 1532 vertical turning and boring mill made by the Kolomensk Heavy Machine Tool Plant. Here lubrication is combined with load relief on the ways. The vee ways of the mill have wedge-shaped pockets. At sufficiently high speeds these pockets provide for hydrodynamic lubrication. In starting and stopping the boring mill, a supplementary lubrication system is turned on in addition to the main system. The supplementary system delivers oil to the pockets at an increased pressure sufficient to raise the table for hydrostatic lubrication.

The design of circular ways does not essentially differ from that of ways for rectilinear motion (see Sec. 10-3).

CHAPTER 11

ELEMENTS OF MACHINE TOOL CONTROL SYSTEMS

11-1. Functions of Control Systems. Requirements Made to Control Systems

The operating features of a machine tool and, in particular, its production capacity, convenience and ease in servicing and its reliability in operation, depend to a great extent upon how well its control system has been designed. In accordance with the nature of the process performed by the machine tool and the consequent construction features it has, the control system can be broken down into a number of trains. Depending upon their functions certain control trains should be independent of the others, while other trains should be interconnected, i.e., interlocked.

As a whole, the control system of a machine tool is often a combination of mechanical, electrical and electronic, hydraulic and pneumatic devices, sometimes almost all of these facilities being employed in a single machine tool.

The degree of automaticity of the control system in up-to-date machine tools ranges from fully automatic controls when, after being started, the machine tool operates with no participation whatsoever of the operator in controlling the machine, to fully manual controls (hand-controlled machine-tools). The general trend in modern machine tool engineering is to automate an ever-increasing number of control operations and to simplify the remaining operations, performed manually, to the maximum possible extent.

Automatic control systems have acquired decisive importance in modern machine tool engineering. They can be classified as centralized (independent or time-sequence control systems) or in-travel (dependent) control systems. In systems of the first type, the control command to the operative member is carried out regardless of its position or whether the preceding command has been carried out. In the in-travel control systems, the operative member carries out a control command after it (the member) has reached a definite position.

There is also an ever-increasing number of programme-controlled machine tools with a positively and automatically accomplished cycle that follows a certain law set up by some interchangeable element or elements. A drum similar to a controller can, for example, be used for this purpose. Copper

strips secured to the surface of the drum close electrical circuits in a definite sequence during drum rotation and thereby function as a control device. Instead of such a drum, punched tape or cards, magnetic tape, tape (film) with an optical record, etc., can be employed. Numerical controls, which are taken up together with other systems of automatic controls of machine tools in Part Six (Vol. 4), are finding wider and wider application.

The problems of control automation have acquired great importance in designing new models of machine tools in connection with the extensive application of high-velocity metal-cutting methods which require that the handling time in controlling a machine tool be reduced to a minimum. This must be taken into consideration in working out the control system for new models of machine tools of all types.

The following requirements are made to control systems:

1. *Safety of control.* To ensure operator safety and health, the control devices should be concentrated and arranged in convenient control zones (within easy reach of the operator) and, if necessary, controls should be duplicated so that the operator need not walk excessively around the machine tool.

Constructions of control systems should be avoided in which certain control devices rotate during machine operation.

Electric push buttons and rotary switch handles should be sunk below the surface of covers, or they should be protected by rings, etc. This requirement does not refer to STOP push buttons.

The following measures are used to prevent accidents that may be the result of shortcomings in the design of a control system or of mistakes made by the operator:

(a) the control members are fixed (locked) in each definite position they occupy in operation;

(b) control mechanisms are interlocked, i.e., linkages are devised between the separate control trains which make it impossible to engage two conflicting motions simultaneously (for example, to engage table feed in a milling machine when the spindle is stationary, or to disengage spindle rotation without disengaging table feed first);

(c) travel limiting devices are provided for positioning motions;

(d) signalling devices are used.

In machining radioactive and toxic materials, it is necessary to apply remote controls (see Sec. 11-7) and to use special safety measures.

2. *Ease and convenience in manipulating* manual control elements.

In laying out the control stations and in arranging the handwheels, levers, handles, cranks, push buttons, knobs and other control elements, it is necessary to take into consideration the physiological factors of human beings. The effort required for operating handwheels or levers of travel mechanisms should not exceed 8 kgf, or 16 kgf if the same traverse motions can be performed mechanically as well. If possible, it is better to take the maximum

effort as 6 or 6.5 kgf, or only 4 to 4.5 kgf if the control operation is performed frequently.

Important factors in control ease and convenience are the size, shape and location of the part of the control element gripped by the hand, and the zone in which the control elements are disposed.

If control elements, travelling during operation together with the unit on which they are mounted, leave the operator and enter a zone in which it is inconvenient to operate them, the control elements for stopping the machine in emergencies should be duplicated together with at least the most important of the other control elements. The most convenient solution of this problem is the use of pendent push-button stations.

3. *Rapid operation of the controls.* The more frequently a control operation is performed, the less the time it should require.

4. *Mnemonic features of the controls*, provided primarily by co-ordinating the direction of hand motion with the direction of travel of the controlled unit of the machine, in accordance with the rules of the USSR Std 9146-59, *Directions of Motions in Machine Tools.* This standard stipulates the directions of motions of the control elements to obtain manual or mechanized travel of the various units in ensuring the relative positions of the workpiece and cutting tool.

The degree to which controls should be mnemonic depends on the number of control elements that the operator must manipulate in operating the machine tool. The greater the number of control elements, the more difficult it is to memorize the controls and more the time required for making change-overs (changing speeds or feeds, engaging or disengaging traverse, etc.). Hence is the tendency to reduce the number of control elements in up-to-date machine tools. The best in this respect is a system in which a single control element is used in making a change-over (see Sec. 11-4) or a single button is pushed in push-button controls.

Another method of reducing the number of control elements is to concentrate several different, but like or related, functions in a single lever or handwheel. The integration of the control of unlike functions is also permissible if the control system is automated to a degree in which mistakes in operation are practically excluded.

5. *Accuracy of the control system.* The accuracy of traverse obtained by various control elements may differ in a wide range. In some cases an accuracy measured in millimetres is sufficient (for instance, in setting the tool head of a planer along the crossrail), while in other cases the required accuracy of travel must be measured in microns (as in positioning in a jig boring machine).

In each separate case, the required accuracy of a control train should be determined on the basis of its application and the function it performs.

11-2. Selecting the Control System and Its Construction

The control system of a machine tool is made up of either independent or interlocked trains. Each of these trains has a definite function in the machine tool and consists of: (a) the control member (element) which receives a command at the required moment in the cycle from the transmitter; (b) elements and transmissions whose purpose is to transmit the command, received by the control member, to the operative member which performs the required control motions, this transmission usually being accompanied by a conversion of the movement of the control member in magnitude and direction, simultaneously with a conversion of the force applied to the control member; and (c) the operative member.

The transmitter of the command may be either the hand or foot of the operator of the machine tool; a dog travelling together with the table, slide, etc.; a cam on the camshaft of an automatic machine tool; a template in the form of a model, master; a graphic template (drawing); punched tape, punched card, magnetic tape, etc.

Commands are transmitted to the operative member of the control train by mechanical elements and transmissions, and electric, hydraulic, electronic and pneumatic apparatus in a great variety of combinations.

The operative member of the control train, accomplishing the required movement of the corresponding part of the machine tool, is in most cases a mechanical element (lever, rack, shifting fork, etc.). Its functions are sometimes performed by oil under pressure or compressed air which acts directly on the part to be moved.

If the machine tool being designed is to be employed for large-lot or mass production, the controls, as a rule, should be fully or almost fully automated; a semiautomatic or automatic machine tool should be designed (see Part Six, Vol. 4).

Applying various types of automating elements and devices, sometimes quite simple ones, the number of control operations can be reduced to a minimum and, in certain cases, a nonautomatic machine tool can be converted into a semiautomatic or automatic one.

The most expedient degree of control automation can be established for each new machine tool being designed by comparing the complication in construction due to this automation and the ensuing increases in labour input and costs and even, in some cases, decrease in operating dependability, on the one hand, with the economical effect achieved by this automation and easier servicing, on the other hand.

Automatic stops for disengaging power feed at the end of the cut or operation are of advantage and sometimes necessary (for example, in drilling and

tapping blind holes in a drill press) though they raise the price of the machine tool, as a rule, only slightly.

The next step also offers considerable difficulties. It involves the selection of the most rational construction for the control system. These difficulties arise because of the great variety of available combinations of mechanical, electric, electronic, hydraulic and pneumatic means that can be employed to solve this problem. For example, a fully automatic control system can be designed using only mechanical elements and transmissions as has been done in many up-to-date automatic screw machines.

A hydraulic control system is to be preferred if the machine tool being designed is to have a hydraulic drive to power the feed or main motion. In such cases, there is no need to install a pump station for accommodating only a control system.

Electric, hydraulic, electrohydraulic and electropneumatic systems are very convenient for remote control of machine tools.

Electric controls are usually the most convenient for machine tools powered by several motors. The potentialities of such controls are very extensive and are continuously increasing. It is possible at the present time, for instance, to synchronize the controls of two tracer-controlled semiautomatic milling machines in such a manner that one will produce a right-hand die at the same time that the other machine is producing a left-hand die to the same template.

The application of pneumatics in a control system is restricted primarily by the requirement that compressed air mains be available in the shop where the machine tool is to be installed.

11-3. Mechanical Control Systems and Their Principal Elements

A system of controls can be devised so that in changing over from one speed to any other speed (or from one rate of feed to any other rate) it is necessary to pass through all the intermediate speeds (or rates of feed). The drawbacks of such consecutive speed changing arrangements are the large time losses in changing speeds, the excessive wear of the gear teeth at their ends in gearboxes with sliding cluster gears or wear of the jaws in gearboxes with jaw clutches.

These drawbacks are absent in a selective speed-changing system, allowing changes from any speed to any other speed (or feed), bypassing all the intermediate speeds (see Sec. 11-6).

The time required for speed changing can be reduced still more if a control system with preselection of speeds (or feeds) is resorted to (see Sec. 11-5).

One shortcoming inherent in all three above-mentioned speed-changing systems is that it is impossible to shift the gears or to engage the clutch

to change the speed if the ends of the gear teeth or the ends of the clutch jaws run up against the teeth or jaws of the mating gears or clutch members. In such cases it is necessary to transmit a slight turning motion to the gearbox shafts either by hand or from an inching push button or, finally, by inching (jogging) engagements of a friction clutch if one is available. All this involves an extra loss of time.

The selection of a type of control system depends mainly upon how often change-overs are to be made and, consequently, how fatiguing the control system will be for the operator, and how large the share of handling time required for changing speeds, feeds, etc., in the total piece (floor-to-floor) time.

Hand Control Elements and Pedals

A great variety of hand control elements are employed in modern machine tools; the most common of these are handwheels with and without spokes, crank handles, handles of control levers, and various knobs which have been standardized in the USSR (machine tool industry standards MH 4-58 through 12-58). Standard control elements should be used in all cases except when special conditions call for elements of specific shapes.

Detachable control elements are, in general, undesirable since they often get lost. Sometimes, however, it becomes necessary to use them, for example, as an extremely simple means of interlocking conflicting control motions.

As previously mentioned, it is very undesirable to allow rotation of handwheels or crank handles during operation of the machine tool, especially during rapid traverse motions when these control elements rotate rapidly and are a hazard to the operator. Handwheels are to be designed so that they automatically become disengaged from the shaft on which they are mounted (for example, by means of a spring) during the whole period of power traverse of the unit they control.

Foot controls are considerably more seldom used in machine tools than hand controls. In most cases, pedals serve for controlling clamping devices, for example pneumatic or electric chucks, since in removing the finished workpiece and loading a new blank, frequently both hands of the operator are occupied.

Transmission from the Control Member to the Operative Member

In the majority of cases the initial motion of the control member is rotary, while the motion of the controlled element or part of the machine tool is more often rectilinear than rotary. Therefore, control trains incorporate

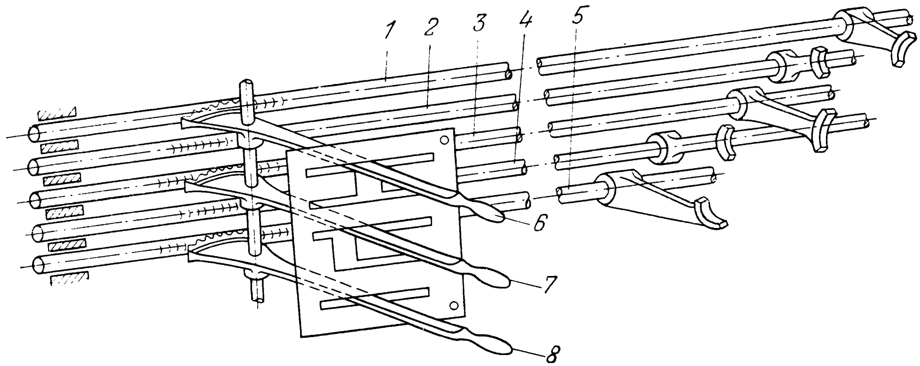


Fig. 155. The use of rack transmissions in the control mechanism of a speed gearbox

all types of mechanical transmission in which rotation or, less frequently, rectilinear motion is converted into rectilinear or rotary motion.

Of the mechanical types of transmission, the most commonly used are levers, racks and screws, while cam, link-motion and Geneva wheel mechanisms are used for single-lever control systems.

The main advantage of a rack transmission is the possibility of arranging the rack at will in the plane of the rack pinion or segment gear meshing with the rack. This enables the part linked to the rack to be moved in any plane. The same rack pinion can be made to mesh with several racks, as is often the case in selective control systems. This may exclude the need for special interlocking elements. Shown in Fig. 155, as an example, are the controls of a speed gearbox which shift five sliding parts along two parallel shafts: two double-cluster gears and three single gears. Shifting is done with three levers, *6*, *7* and *8*, mounted on a common axle, by means of segment gears meshing with racks cut on five control rods, *1*, *2*, *3*, *4* and *5*, carrying forks, one for each of the shifted elements of the gearbox.

A drawback of this solution is the lack of sufficient convenience in controlling the gearbox because of the many levers.

Transmission with a screw and nut is especially convenient for accurate motions. In combination with high-ratio reduction gearing, a screw and nut can be employed for making very small hand-operated motions, measurable with a micrometer, similar to those required for intermittent infeed in a grinding machine. An advantage of a screw drive is that it enables a large force to be produced at the end of the control train, one required, for instance, for traversing a heavy unit of the machine tool without introducing intermediate transmissions into this train. Of importance in some cases is the fact that a rack drive enables a unit to be traversed more rapidly than

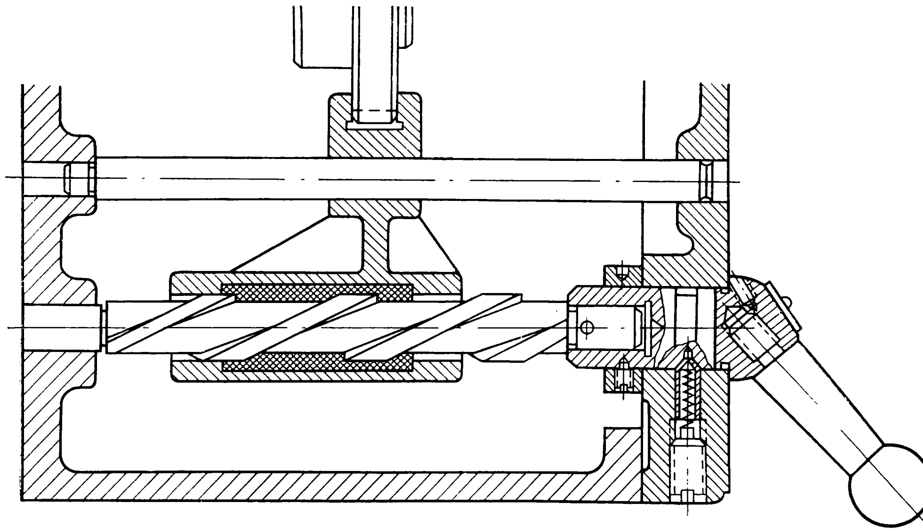


Fig. 156. Components controlling a sliding cluster gear in the feed gearbox of a planer

a screw drive. This difference between the two types of transmission disappears if a high-lead screw (Fig. 156) is employed.

Transmissions with Geneva wheels, pin wheels, cams and incomplete gears prove convenient in controlling several sliding cluster gears, clutches, etc., from a single handwheel or lever (see below).

Widely employed for changing spindle speeds or feeds is a simple lever mechanism in the form of a rod with a fork. Mounted on the outside end of the rod is a handwheel or lever. The fork is linked to the part being shifted only axially and does not impede its rotation. Upon turning the lever the fork shifts the clutch or cluster gear along the shaft to the required position. If the element is to be shifted over a comparatively long distance, it will be necessary to provide guides for the fork, for example, in the form of strips, round rods or a spline shaft to avoid misalignment of the fork.

11-4. Multiple-Lever and Single-Lever (Single-Handle) Control Systems

The control trains for the parts of the same unit can be made independent of each other. Such a solution usually leads to an unwieldy multiple-lever

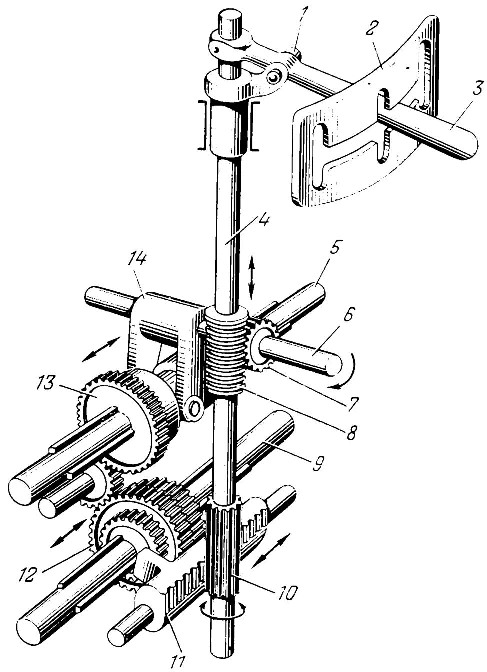


Fig. 157. An example of a single-lever control system

(or multiple-handle) control system which is inconvenient in operation, inefficient as to time lost in controlling the machine and fatiguing for the operator.

Single-lever (single-handle) systems are much more efficient. Here each unit is controlled by one or two hand control members. Such systems represent one of the typical trends in the design of modern machine tools in which manual control still plays a significant role. The mechanisms of single-lever controls are frequently quite complicated and expensive. Hence, in designing a new model it is necessary to compare versions of both systems of control and to judge to what extent a more complicated and expensive construction is justified by the operational advantages and economic assets of a single-lever control system.

If the machining time of an operation constitutes many hours, an economy of seconds or even minutes in carrying out the hand control operations is of no significance. In such cases, a single-lever control system may be justi-

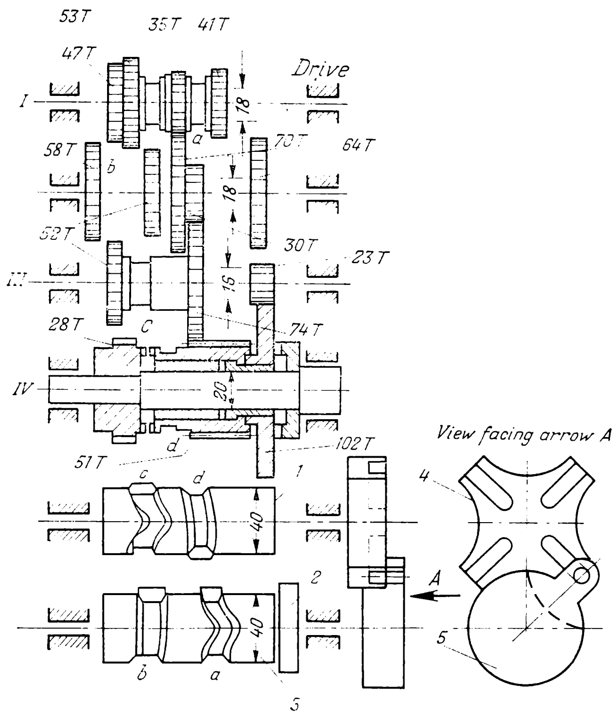


Fig. 158. Single-lever control system with control drums and a Geneva wheel drive

fied by the effort to avoid the possibilities of mistakes in setting up or operating the machine that can lead to rejection of the workpiece. On the contrary, a single-lever system is to be preferred in all cases when the operator must manipulate the hand control devices comparatively often as in the operation of medium- and small-size machine tools.

The most widely applied single-lever control systems can be divided into two main groups:

1. Single-lever control systems with permanent linkages between the controlling members and the parts being controlled. All motions of these parts are accomplished as a result of the selected structure and construction of the control trains. Widely employed in such trains are cylinder (drum) and plate cams, link-motion drives, Geneva wheel mechanisms, as well as hydraulic, pneumatic and electrohydraulic devices.

Spindle speed step	Transmissions in operation				Cluster gear positions a	b	c	d
n_1	35/70	—	30/74	23/102	L.H.	0	R.H.	R.H.
n_2	41/64	—	30/74	23/102	R.H.	0	"	
n_3	—	47/58	30/74	23/102	0	L.H.		
n_4	—	53/52	30/74	23/102	0	R.H.	"	
n_5	35/70	—	58/52	23/102	L.H.	0	L.H.	
n_6	41/64	—	58/52	23/102	R.H.	0	"	
n_7	—	47/58	58/52	23/102	0	L.H.	"	"
n_8	—	53/52	58/52	23/102	0	R.H.	"	"
n_9	35/70	—	30/74	74/51	L.H.	0	R.H.	L.H.
n_{10}	41/64	—	30/74	74/51	R.H.	0		"
n_{11}	—	47/58	30/74	74/51	0	L.H.		"
n_{12}	—	53/52	30/74	74/51	0	R.H.		
n_{13}	35/70	—	58/52	74/51	L.H.	0	L.H.	"
n_{14}	41/64	—	58/52	74/51	R.H.	0	"	"
n_{15}	—	47/58	58/52	74/51	0	L.H.	"	"
n_{16}	—	53/52	58/52	74/51	0	R.H.	"	"

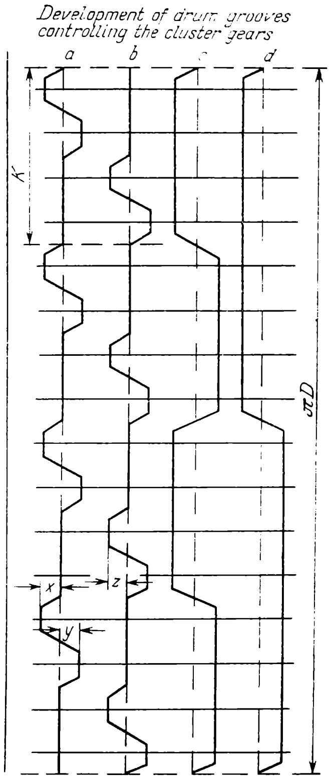


Fig. 159. Development of the cam grooves on drums *1* and *3* of the control mechanism illustrated in Fig. 158

2. Single-lever control systems in which the same controlling member can be linked to several different control trains. The controlling member in this case is a lever or handwheel which can be shifted along its shaft, or a joy-stick type of lever with a constant centre of swivel, etc.

The principles of single-lever control system design are explained by the following examples.

In the construction shown in Fig. 157, lever *3* can be turned both in a horizontal plane together with shaft *4* and in a vertical plane about pin *1*. When the lever is turned in a horizontal plane, spool gear *10*, integral with shaft *4*, moves plunger and rack *11* with its fork, thereby shifting triple-cluster

gear *12* to the right or left along shaft *9*. If lever *3* is turned in a vertical plane, shaft *4* is moved upward or downward and circular rack *8* turns gear *7* with shaft *6* on which it is secured. Fork *14*, keyed on shaft *6*, then shifts double-cluster gear *13* along shaft *5*. The six slots in gate *2* correspond to the six steps of speed obtained by all possible combinations of engagement of the triple- and double-cluster gears. When lever *3* is in the central position (not in any of the slots) the two cluster gears are in their neutral positions.

Figure 158 illustrates schematically a single-lever system for controlling 16 spindle speeds of a milling machine. Spindle speeds are changed by shifting four sliding double-cluster gears, a, b, c and d. To construct the developments of the curves on drums *1* and *3* (the first controlling cluster gears c and d, and the second, cluster gears a and b), it is sufficient to have a structural diagram of the speed gearbox which clearly shows the sequence of engagements. Using such a diagram, no difficulty is encountered in plotting the development of the curves on the control drums, as has been done for the given gearbox in Fig. 159.

Dimensions x, y, z, etc., and the width and shape of the grooves are determined in accordance with the length of shift of the cluster gears, diameter of the follower rollers that enter the grooves, etc.

In the arrangement shown in Fig. 158, the shafts of the two drums are linked through a Geneva wheel transmission *5* and *4*, with a four-slot wheel. Thus, drum *1*, controlling cluster gears c and d, makes one full revolution to every four revolutions of drum *3* which controls cluster gears a and b. This corresponds to the structural formula of the gearbox, which is $16 = 4 \times 2 \times 2$. For this reason, there is only one right and one left working position (section K of the development in Fig. 159) in the groove on drum *3* for cluster gears a and b. The same is true of the groove on drum *1* for cluster gear d (Fig. 158) while the groove on drum *1* for cluster gear c has two right and two left positions (Fig. 159).

The device is locked in the working positions by means of disk *2*.

In comparison with cylinder cams (drums), plate cams with a positive-return feature (face cams) have the advantage of occupying less space due to their small thickness and the possibility of arranging curves on both sides of each disk. The use of face cams in a single-lever system for controlling the spindle speeds of a milling machine is illustrated in Fig. 160. As is evident in the kinematic diagram, the spindle has $4 \times 2 \times 1 \times 2 = 16$ speed steps. The speed gearbox is similar to that shown in Fig. 158 and its controls differ only in that two disks replace the two drums. Disk *1* controls two double-cluster gears c and d of the two extension groups of transmissions for which purpose it (the disk) has two grooves (face cams), one on each side. Disk *2* has only one groove (face cam) for the two double-

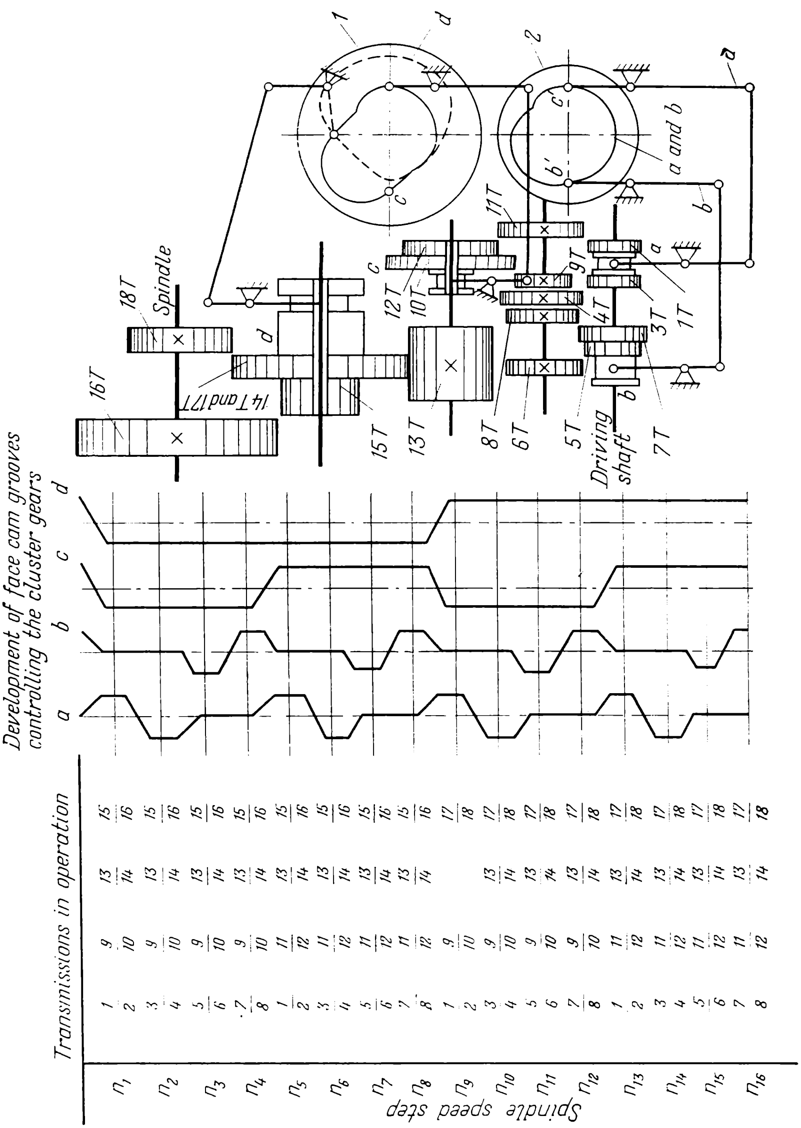


Fig. 160. Single-lever control system with disk (face) cams and lever transmissions

cluster gears a and b. Roller followers a' and b', controlling these cluster gears, are arranged in this groove at an angle of 180° from each other.

If the control of all the cluster gears was combined in a single face cam, rollers a' and b' would be displaced from each other by an angle of $\frac{360°}{16}\,2 = 45°$, as can be seen by comparing the developments of the two grooves for cluster gears a and b. Consequently, in the construction selected for the control mechanism, disk *2* should make $\frac{180°}{45°} = 4$ revolutions per revolution of disk *1*. In this respect this mechanism does not differ in construction from the one shown in Fig. 158. This is accomplished by providing the corresponding drive between disk *2*, directly linked to the crank handle, and disk *1*. The set-up spindle speed is indicated on a dial by an arrow linked to the disks.

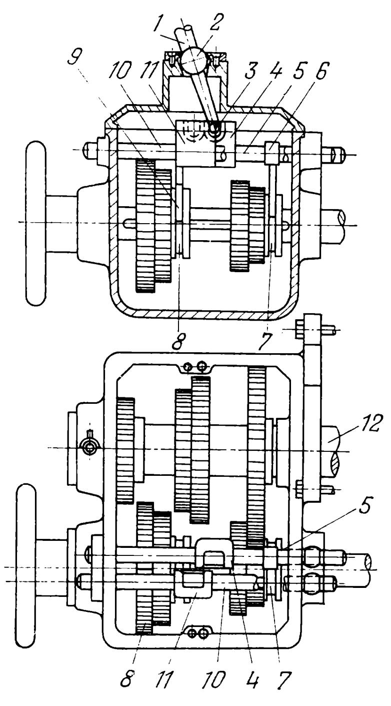


Fig. 161. Four-position joy-stick type lever

A great number of parts in the mechanisms of a machine tool can be shifted by means of a single lever or single handwheel. This enables a large number of main drive speeds, rates of feed, rapid traverse and positioning motions to be controlled. Making use of electric, hydraulic and/or pneumatic facilities, especially convenient for remote control, the lever or handwheel of the single-lever control system can be separated from the machine tool proper by arranging it on a pendent or portable control station.

The choice of the combination of mechanical and other means, most expedient from the point of view of operational features and producibility, should be conditioned, in each definite case of designing a new model, by the increase in production capacity attainable by the use of a single-lever control system, on the one hand, and the increase in the cost of the machine tool resulting from a more complicated control system, on the other hand. In assessing the merits of the various feasible versions of single-lever controls, one should not overlook an essential shortcoming of many versions—the

necessity in changing speeds or feeds for passing through all the intermediate steps. This leads, not only to unproductive losses of time, but also to increased wear of the components of the control system. Hence, if speeds or feeds are to be changed frequently, such versions are especially undesirable.

Joy-stick type levers. These levers, which can be turned in two or several planes, have the advantage that they need not be shifted through all the intermediate positions. The use of such a lever is demonstrated in Fig. 161. Two double-cluster gears *7* and *8*, meshing with four gears on driven shaft *12*, can be shifted by forks *6* and *9*, secured on the shifting rods *5* and *10*. The two rods, together with their forks, are shifted by means of a single lever *1* mounted with its ball joint *2* in the cover of the gearbox. Ball tip *3*, at the end of the lever, can be entered into the recess in block *4* or *11*. When lever *1* is turned, it will shift the given block together with its rod along the rod axis, thereby shifting the corresponding cluster gear.

When shifting rods *5* and *10* are in their middle position (in which case the two cluster gears are in their neutral positions), the recesses in blocks *4* and *11* are opposite each other and ball tip *3* can be entered into either recess. When one of the blocks has been shifted to its working position, it is impossible to enter ball tip *3* into the other recess.

A single joy-stick type lever can be employed to control a large number of speeds or feeds by engaging it to different control trains.

11-5. Control Systems with Preselection of Speeds or Feeds

The time lost in change-overs can be reduced if the control system is designed so that the greater part of the manipulations needed for this purpose are performed while the machine tool is in operation but without altering the speeds or feeds set up for the present operation. At the end of the operation, the speed (or feed) is rapidly changed by a single motion of a lever or by simply pressing a push button. Such control systems are called preselective because they enable the speed or feed for the next operation to be selected during the current operation. These systems can be economically justified in machine tools whose operation requires comparatively frequent changes in the speeds or feeds (for instance, turret lathes). If the machining time for the various operations is large (for example, in machining heavy workpieces in large machine tools), it proves impossible to economically substantiate the use of a preselective control system. The decision to apply such a system should be based in each individual case upon comprehensive engineering and economical calculations.

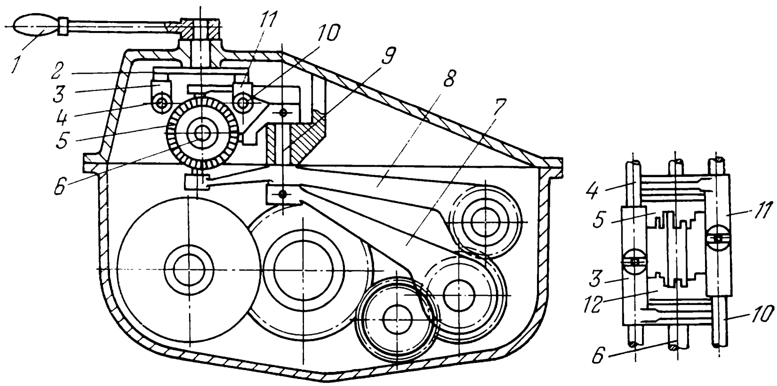


Fig. 162. Control system with speed preselection

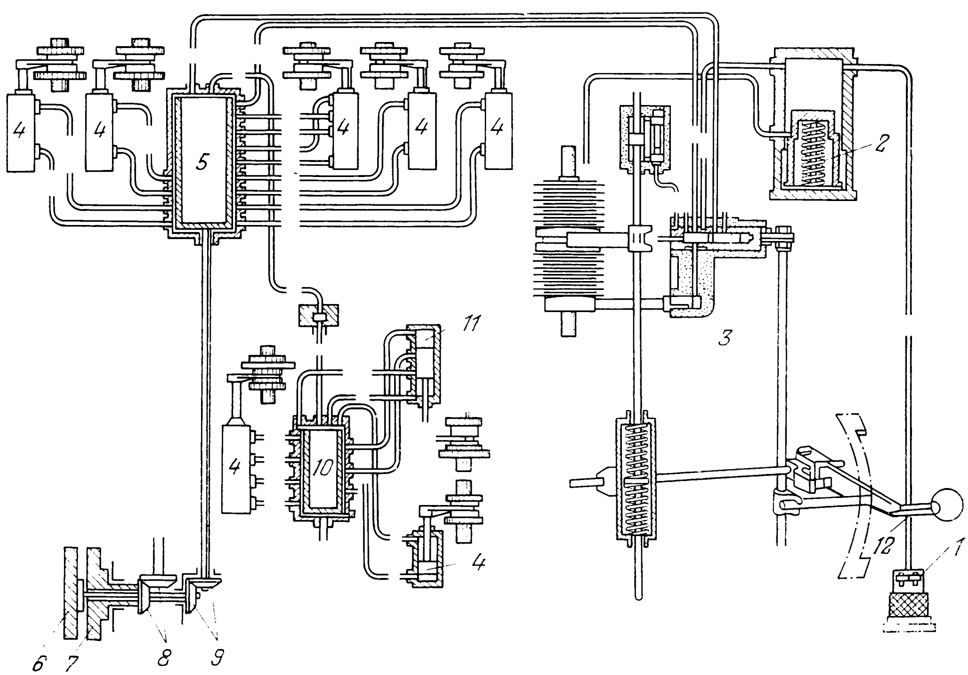


Fig. 163. Hydraulic preselection system for controlling spindle speeds

Quite a large number of different systems of speed preselection controls are being used in up-to-date machine tools. These systems may differ substantially as to their constructions but common to all of them are the principle of disengaging the control train during the time preparations are being made for engaging the next speed, and the use of contoured cams for shifting, similar to the cams or drums of a single-lever control system.

The principle of a speed preselection system is illustrated in Fig. 162. First of all lever *1* is turned. At this, part *2* shifts forks *3* and *11* along rods *4* and *10* in opposite directions. These forks enter annular grooves of contoured cams *5* and *12*. Then shaft *6* is turned with a handwheel. The cams are mounted on splines of this shaft. The speed is changed by shifting lever *1* back in the opposite direction. This disengages the main friction clutch, the speed gearbox shafts are slowed down and cams *5* and *12*, brought together again, actuate with their lobes the pins of levers *7* and *8*. This turns levers *7* and *8* about the axis of pin *9* so that the forks at the ends of the levers shift the corresponding cluster gears to their new positions.

Shown in Fig. 163 as a second example is the schematic diagram of a hydraulic preselection system for controlling the spindle speeds and feeds of a radial drill. The system operates as follows.

Pump *1* delivers oil to accumulator *2*. When the accumulator is filled with the necessary volume of oil, a port opens through which oil under a pressure of 10 to 12 kgf per sq cm passes for lubricating the bearings and the gears of the drill head.

Preparations are made for shifting the cluster gears in the speed and feed gearboxes by turning handwheels *6* and *7*. Through bevel gearing units *8* and *9*, these handwheels turn the internal sleeves of the preselector valves *5* and *10*. At this the pressure chambers of these valves are connected to the upper and lower ends of the two-position cylinders *4* and the three-position cylinder *11* whose pistons are linked rigidly with the levers of forks that shift the cluster gears. As long as main valve *3* is closed and the ends of the actuating cylinders are not yet under pressure, all the cluster gears remain in their previous positions. The cluster gears are shifted as soon as valve *3* is connected to the hydraulic control system by shifting lever *12* which also engages the multiple-disk clutch of the drill drive.

A comparison of existing preselection control systems leads to the conclusion that their further development will be toward the application of hydraulic, pneumatic and electric facilities in them, simplifying, at the same time, their mechanical components.

11-6. Selective Speed and Feed Changing Systems

The operational shortcomings of consecutive speed-changing systems were mentioned on p. 241 together with the advantages of selective systems. The more the speeds or feeds are changed, i.e., the more the number of steps of spindle speed or feed, the more significant these advantages become. The principle of operation of a selective speed-changing mechanism is explained by Figs. 164 through 168.

Figure 164 is an elementary diagram of a single-lever selective speed-changing mechanism designed in the Sverdlov Machine Tool Plant for application in the horizontal boring machines of this plant. To engage one of the four available speeds of driven shaft *7*, selector disk *2* is pulled by means of lever *1* toward the operator, thereby moving it away from the pusher racks *3* which mesh in pairs with pinions *11*. Then the disk is turned to the required position, in which an arrow or other index will point to the required speed (or feed) value on a dial or circular scale, and pushed forward as far as it will go. At this the selector disk pushes forward the corresponding pair of pusher racks (the other pair being opposite holes in the disk) and the forks of levers *5* and *10* shift the double-cluster gears *6* and *9* to the required positions. Electric element *4* serves here to inch drive motor *8*, facilitating engagement as the gears slide into mesh.

The selective speed-changing mechanism shown schematically in Fig. 165 also has a single selector disk. It differs from the preceding mechanism only in that it has a greater number of rack pushers by means of which four double-cluster gears, *1*, *2*, *3* and *4*, are shifted to obtain $4 \times 2 \times 2 = 16$ speeds

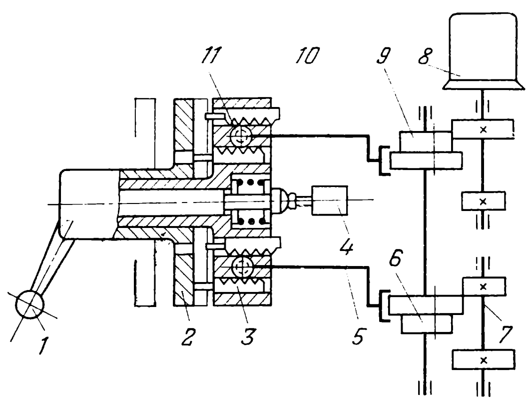


Fig. 164. Elementary diagram of the single-lever selective speed-changing mechanism for controlling four speeds in the model 262Г horizontal boring machine manufactured by the Sverdlov Machine Tool Plant of Leningrad

The principal geometric dimensions of such mechanisms can be determined from quite evident relationships, using the notation accepted in Figs. 165, 166 and 167.

The distance between the axes of the pairs of rack pushers is

$$C = 2r + d - 2m \tag{215}$$

where m is the module of rack pinions z_1, z_2, z_3 and z_4 (Fig. 165).

The concentric circles on which the centres of the holes for the pins of the rack pushers are located have a diameter

$$D_i = \frac{C}{\sin\frac{\beta}{2}} = \frac{C}{\sin\frac{k\alpha}{2}} \tag{216}$$

where k = a whole number

$\alpha = \frac{360°}{n}$

n = number of different engagements.

Hence

$$D_i = \frac{C}{\sin\frac{k\times 180°}{n}} \tag{217}$$

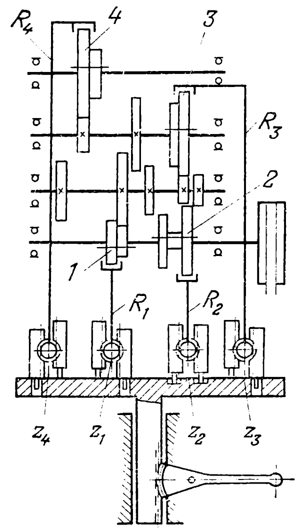


Fig. 165. Elementary diagram of a single-disk selective speed-changing mechanism for 16 speeds

The length of travel of the rack pushers depends upon the width b of the toothed rims on the cluster gears and the ratio $\frac{R_i}{r}$, where R_i is the radius of the lever that shifts the corresponding cluster gear. As a rule $R_i = (3 \text{ to } 5)r$.

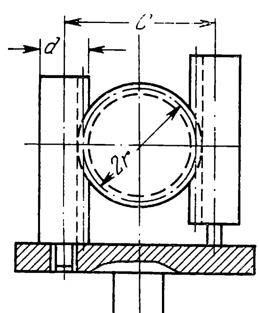


Fig. 166. Diagram for determining the geometric dimensions of elements of the mechanism illustrated in Fig. 165

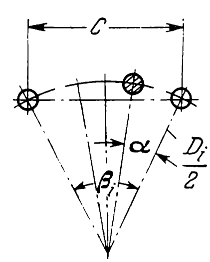


Fig. 167. Diagram for determining the geometric dimensions of elements of the mechanism illustrated in Fig. 165

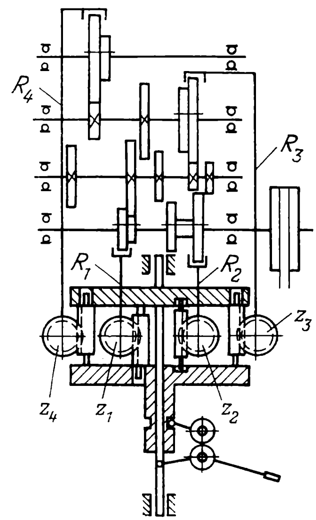


Fig. 168. Elementary diagram of a two-disk selective speed-changing mechanism for 16 speeds

The selective speed-changing mechanism shown in Fig. 168 differs from those described above in that it has two selector disks instead of one. The other differences of this mechanism from the one shown in Fig. 165 require no further explanations.

Though they possess an important operational advantage—the possibility of changing from any speed to any other speed without passing through all the intermediate speeds—these selective mechanisms also have their shortcomings. In construction and manufacture they are quite complicated, cluster gears mounted on a single shaft of the gearbox must be provided with an interlocking device avoiding simultaneous engagement of two cluster gears, etc. A two-disk selective mechanism requires less space than a single-disk mechanism; otherwise they are practically equivalent. A more detailed analysis of selective control mechanisms and an improved version of them are to be found in an article *Selective Speed-Changing Mechanisms* by M. Moldavsky (Stanki i Instrumenty, No. 11, 1959).

An analysis shows that the number of repeated disengagements and engagements of interlocked cluster gears, depending upon the construction of the selective mechanism and the interlocking method, can be quite considerable and lead, not only to excessive time losses in changing speeds, but also to excess wear of the ends of the teeth on the sliding gears. In this respect, selector mechanisms with rocker arms (used, for example, in the model 679 drill flute milling machine) or with rotary pin-type pushers (as in the ram-head milling machines, models 675 and 676) are more efficient than mechanisms with rack pushers since they do not require repeated, kinematically unnecessary, engagements.

11-7. Remote Controls

Remote controls, extensively applied in up-to-date machine tools, enable the operator to perform the greater part of the control operations, being himself at a more or less considerable distance from the machine tool units being controlled. Such control systems are convenient in many cases, and are almost indispensable for heavy and especially for unique machine tools. Machine tools intended for machining materials possessing natural or arti-

ficial radioactivity (radioisotopes) are equipped with remote control facilities for all operations, from the clamping of the blank in the machine to the removal of the finished or semifinished part. This is all the more necessary because machine tools for this purpose are frequently installed in a separate room, isolated from the operator, or they operate at the bottom of a deep well filled with water.

Remote controls are also used on machine tools intended for machining blanks of beryllium which constitutes a hazard due to its toxicity.

In large-size machine tools, remote controls cover a greater or fewer number of operations depending upon the frontal size of the machine, its height and sometimes its width, and upon its construction—the arrangement of the units, size and weight of the blanks and the frequency with which control operations are to be performed.

In accordance with the location of the control station on which the control members of the machine tool are concentrated and also from other considerations, various systems of remote controls can be used. These systems include electromechanical, electrohydraulic and others.

The centralized remote control station is most often designed in the form of a pendent (or duplicate pendents for very large machines) on which electric push buttons and sometimes certain control levers are mounted. In most cases, such pendents carry START and STOP push buttons for the main drive, and similar pairs of push buttons for certain units (for instance, the table of a vertical boring mill, crossrails, heavy slides, clamping devices, gearboxes, etc.).

The layouts and constructions of remote control systems are distinguished for the great variety of means employed, elements of systems applied and their combinations. The very principle of remote control predetermines an especially wide application of electric and hydraulic facilities in such systems, and to a lesser degree, pneumatic devices. Speeds and feeds can be conveniently changed in gearboxes by means of electromagnetic friction clutches enabling complete remote controls of main drive speeds, working feeds, rapid traverse movements, reversals of the machine units and, in some cases, preselection of the next speed (or feed). The use of electromagnetic clutches, solenoids, individual electric motors for certain units, etc., enables the functions of the operator to be reduced mainly to pressing push buttons and turning handles at the control station.

Industrial (closed-circuit) television has been used to some extent for cases when the control station is located so that direct visual observation of the machining operation, condition of the cutting tools, and readings of the instruments is difficult or impossible. Such an installation has been applied, for instance, on the heavy vertical turning and boring mill, model КУ-65 (with a table 6.5 m in diameter). The television camera of this outfit can be traversed up and down in the vertical direction and swivelled in

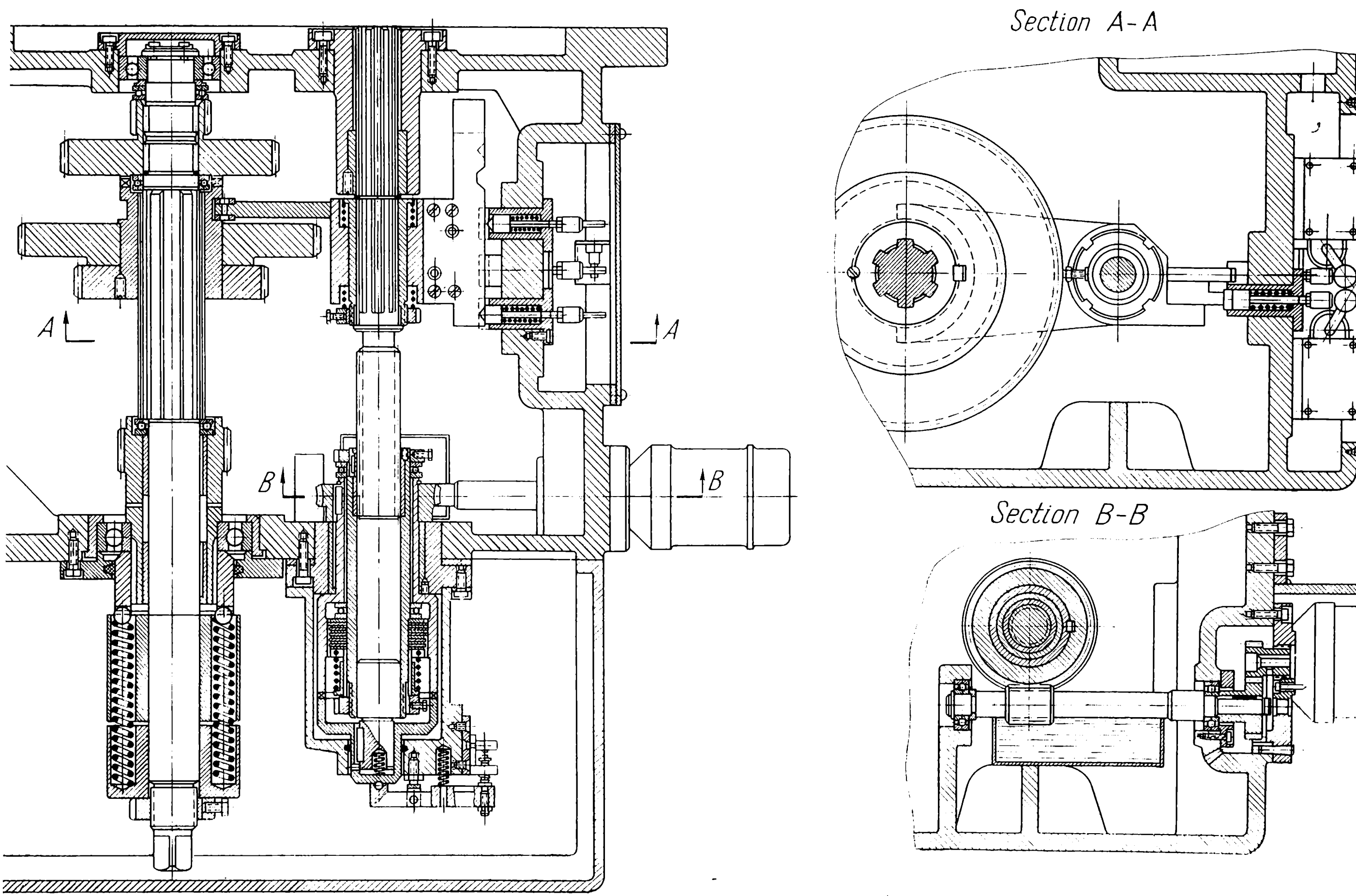


Fig. 169. Mechanism for remote control of the horizontal feeds in the model 7M386 heavy shaper

a horizontal plane. The camera is remote controlled, having a centralized control desk.

Shown in Fig. 169 is an example of the mechanism used for the remote changing of horizontal feeds in a heavy shaper. The electric motor of this mechanism is switched on from the control desk. Through spur and worm gearing and a multiple-disk friction clutch and nut, translatory motion is transmitted to the screw which is rigidly linked to the shifting fork of a cluster gear. When the cluster gear reaches the required position, the fork is automatically locked by an index, the corresponding limit switch is operated and the motor is switched off.

11-8. Safety Devices of Machine Tools

In designing a new machine tool, serious attention must be given to the protection of the operator and servicing hands against accidents and excessively large physical strain, as well as protection of the machine, cutting tool and workpiece against damage that may occur from various causes. Protection of the machine tool against accidents and breakdowns is an especially vital factor if it is to be built into an automatic transfer line.

Safety devices should operate automatically.

Devices for protecting the life and health of the operator are of prime importance. The construction of devices for this purpose is considered in works on safety engineering and man analysis charts.

Devices for protecting the machine tool and cutting tool against breakage or damage, and the workpiece from being spoiled can be classified into three main groups:

1. *Interlocking devices* (*interlocks*) which should ensure: (a) that two or more pairs of gears cannot be engaged simultaneously in a single transmission group since this leads to inevitable breakage of the gears, shafts and other parts (see pp. 260 to 262); (b) that two conflicting motions cannot be engaged simultaneously; and (c) that certain control operations cannot be performed except in a definite sequence and, in some cases, with definite time intervals between them.

2. *Travel-limiting devices* which may have two kinds of purposes in machine tools:

(a) One type of device stops the motion of the travelling unit of the machine tool when it reaches the permissible extreme positions before running off the ways or up against stationary parts of the machine, the cutting tools or the workpiece. Such devices, which operate at the permanent extreme points along the line of travel of a movable part of the machine tool, can be called extreme-position limiting devices.

(b) Other limiting devices are intended for disengaging or switching over the motion of a travelling unit at points along the line of travel that are established in setting up the machine tool. According to their purpose such devices can be called size-maintenance or processing or adjustable limiting devices to distinguish them from the first type which usually occupy a fixed position on the machine tool.

Thus, the main function of the extreme-position limiting devices is to protect the machine against breakdowns, while the function of size-maintenance limiting devices is to ensure that the machining operation is done so that the workpiece acquires the required dimensions, i.e., to prevent spoilage of the work due to size errors.

Size-maintenance limiting devices are extensively used, for example, in turning, grinding or milling up to a shoulder, in plunge-cut grinding, in machining blind holes, etc.

3. *Overstress protection devices* protect the machine tool against excessive loads which may result in the development of such high stresses in the material of certain parts of the machine tool that they lead to breakage or permanent set of the parts, or to stalling of the electric drive motor. Also impermissible are loads resulting in such high elastic strain of certain parts that the mechanisms of the machine tool cannot operate normally.

An excessive increase in the temperature of friction surfaces may also have grave consequences, especially in the case of bearings or slideways. Such overheating may be due either to overloading of the machine tool or to troubles or failure in the lubrication system. Overheating or mechanical overstressing usually lead to seizing of a bearing, scoring on slideways, etc.

The protection devices mentioned above can function satisfactorily only if they operate completely automatically.

Interlocking Devices

The problem of interlocking transmissions can be successfully solved if an efficient control mechanism is devised. In a single-lever control system, simultaneous engagement of two different spindle speeds or two different rates of feed is impossible. Likewise, in a multiple-lever system, no special interlocking elements will be required if the parts of the mechanism, whose simultaneous engagement could lead to a breakdown, are linked to a single hand-control member.

If the control system is designed so that erroneous engagements, dangerous to the machine tool, are possible, then the corresponding control members should be interlocked. The same refers to the control of kinematic trains of conflicting motions, or motions which must take place in a definite sequence.

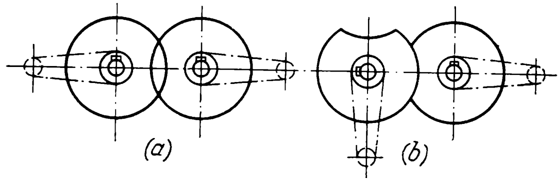


Fig. 170. Interlocking parallel shafts

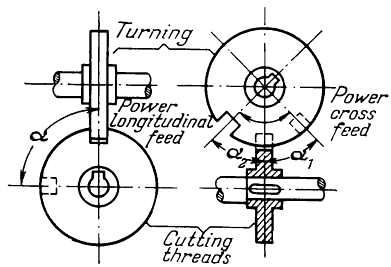


Fig. 171. Interlocking perpendicular shafts

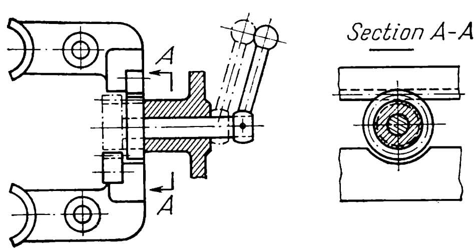


Fig. 172. Interlocking control components travelling parallel to each other

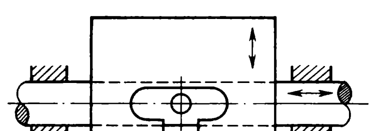


Fig. 173. Interlocking control components travelling perpendicular to each other

Interlocking may be accomplished by various means: mechanical, electric, hydraulic or their combinations.

1. *Interlocking parallel shafts.* Interlocking components are secured on the shafts on which control members are mounted. These components may be in the form, for example, of disks or sectors with concave recesses as shown in Fig. 170. In the position shown in Fig. 170*a*, either crank handle can be freely manipulated; the right-hand shaft is locked in the position shown in Fig. 170*b*.

2. *Interlocking perpendicular shafts.* Shafts perpendicular to each other can be interlocked with similar elements (in Fig. 171 the lower shaft is locked).

3. *Components with rectilinear motion* can be interlocked as shown schematically in Figs. 172 and 173 for the two principal cases.

A definite sequence of control operations, with or without specified time intervals between them, can be maintained by means of electric or hydraulic apparatus or a combination of the two.

Electro-, hydro- and electrohydromechanical facilities are finding wider and wider application for interlocking purposes in new models of machine tools. Their constructions are so numerous and diverse that they cannot be taken up here in greater detail.

Travel-Limiting Devices

The choice of the principle and construction of an automatic travel-limiting device depends upon the functions of this device (extreme-position or size-maintenance limiting device) and the required accuracy with which the travel of the movable unit should be limited.

Extreme-position limiting devices are adjusted in such a manner that the travelling part of the machine tool does not reach the dangerous end position by 3 or 4 mm. Hence, an accuracy of ± 0.5 to 1 mm, and sometimes several millimetres, is sufficient for an extreme-position limiting device.

If the travelling unit is to be powered by an individual electric motor, the unit can be stopped most simply at its extreme positions by means of electric ordinary or momentary-contact button limit switches. An ordinary limit switch with no supplementary devices can stop a travelling unit with an accuracy of 0.5-1 mm.

In cases in which no individual motor powers the part of the machine tool that is to be automatically stopped at the extreme points of travel, it is stopped at these points by disengaging the kinematic train. This can be accomplished by any of the devices used for size-maintenance travel limiting.

As a rule, size-maintenance limiting devices should limit the travel with considerably higher accuracy than the extreme-position type, since the accuracy of the machined workpiece depends upon the former.

The accuracy with which travel is limited also depends upon whether the corresponding part of the machine tool is to be reversed immediately after being stopped. If it is, the accuracy with which travel is limited by a limit switch is much higher since in the process of reversal the electric motor is slowed down rapidly by plugging.

Limit switches can stop travelling units with an accuracy of ± 0.02 or 0.03 mm. This is sufficiently accurate for many cases. When higher accuracy is required, up to ± 0.001 mm, it will be necessary to resort to mechanical or combined electromechanical or electrohydromechanical devices.

Mechanical systems of precise travel-limiting controls are based on the following principle: at a definite point in its line of travel the part of the machine tool whose motion is to be limited meets a rigid (positive) stop secured to some stationary part of the machine tool. As a result the resistance to further motion increases to a point where the kinematic train of the drive to the travelling part is automatically disengaged. This may be accomplished by various means, the most widely used being shown schematically in Figs. 174 and 175. In the diagram in Fig. 174*a*, slide *2* is stopped when it reaches positive stop *1*, and friction clutch *3* begins to slip. This continues until the slide is withdrawn from the stop, for instance by reversing the electric motor. In Fig. 174*b*, ratchet-tooth clutch *3* is used in place of the friction clutch.

A travel-limiting arrangement based on a dropping worm is shown schematically in Fig. 175*a*. The feed motion is transmitted to the slide by feed rod *2* through gearing $\frac{z_1}{z_2}$, shaft *3*, universal joint and shaft *4* on which worm *5* is freely mounted. The worm is linked to shaft *4* by means of an overload clutch *6*. When the slide runs up against positive stop *1*, worm wheel *9* and worm *5* stop rotating, the torque on shaft *4* increases and the overload clutch disengages. At this, the movable part of the clutch shifts to the right, turning lever system *8* in the direction of the arrows and cradle *7* together with the dropping worm swing downward by gravity, thereby disengaging the elements of the worm gearing.

When worm wheel *2*, in the arrangement shown schematically in Fig. 175*b*, stops rotating because the slide has run up against a positive stop, worm *1* continues to rotate, advancing to the right by screw action in the teeth of the worm wheel and turning angle lever *5* counterclockwise. Then, under the action of spring *3*, clutch *4* is instantaneously disengaged.

The accuracy with which travel is limited by mechanisms designed according to the diagrams in Figs. 174 and 175, or by similar arrangements operating on the same principles, depends upon the rigidity of the elements and of the mechanism as a whole.

Of the described versions, the one incorporating a dropping worm is the best since its operation involves the disengagement of components travelling

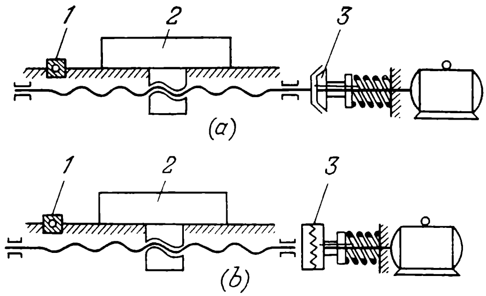


Fig. 174. Travel-limiting controls with a positive stop and slipping clutches

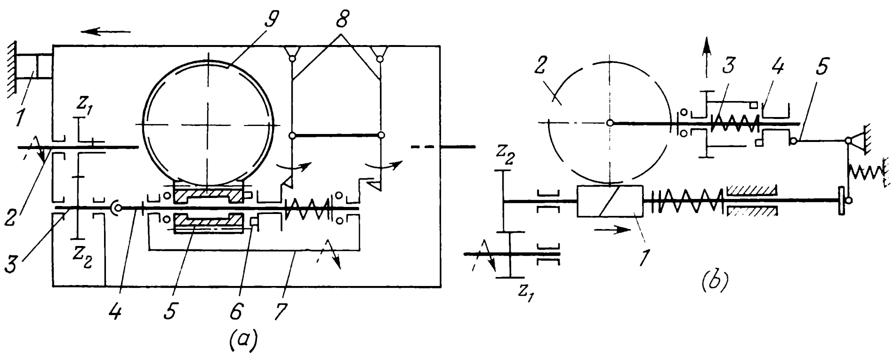


Fig. 175. Travel-limiting controls with a positive stop and a dropping or shifting worm

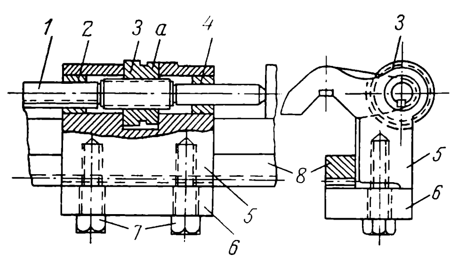


Fig. 176. Securing a positive stop on the bed

at low speeds; they have little inertia and consequently only small overtravel due to inertia.

Travel-limiting accuracy, attained with dropping worms or dropping levers and jaw clutches, is about 0.02 or 0.03 mm for idling and only 0.2 or 0.15 mm under load.

The highest travel-limiting accuracy attainable with a positive stop is ± 0.01 mm, but frequently it is not better than 0.05 mm (this accuracy depends on the mass of the unit whose travel is limited, its speed of travel and the coefficient of friction of the unit on its ways). Higher accuracy can be attained only under exceptionally favourable conditions.

The drawbacks of purely mechanical travel-limiting devices can be eliminated without sacrificing the accuracy of positive stops by using combined electromechanical arrangements. These may be of a great variety of different schemes and constructions. Some devices employ electromagnetic or thermal relays which switch off the drive motor upon a sudden increase in current at the moment that the travelling parts of the machine tool run up against the positive stop. In others, a limit switch is operated simultaneously with the disengagement of the clutch, dropping or shifting worm, to switch off the drive motor through a contactor.

The stop should be of a construction that enables it to be firmly and dependably secured. The stop itself should be wear resistant and rigid since its deformation will reduce to naught all other design measures intended to ensure a high travel-limiting accuracy. The method of fastening the stops depends upon the construction of the parts of the machine tool on which they are mounted. Frequently T-slots in the bed, slides, tables, etc., are used for this purpose.

One of the pair of mating stops should be equipped with a micrometric screw or collar or other similar part enabling the length of travel or arc of swivel to be set up to greater accuracy.

In Fig. 176, stop *5* is secured on the bed with strap clamp *6* which has teeth that enter the tooth spaces of rack *8*. Two screws *7* draw together the clamp and stop, thereby fastening the stop rigidly to the bed. Micrometric screw *1* is supported in bushings *2* and *4*. It is set up by turning nut *3* having a scale engraved on surface *a*. Screw *1* is restrained against rotation by a key which fits a keyway in bushing *2*.

In the operation of machine tools having slides or heads carrying several cutting tools, the lengths of travel of these tools generally differ and therefore each tool requires a separate stop. If there is a great difference in the lengths of travel, a common stop is sometimes designed so that it can be rapidly set up in different positions.

In horizontal and vertical turret lathes, the length of travel of each tool in the turret and the cross slide or side head is limited by a separate stop. Each group of stops, associated with one slide (or head) is arranged on a spool

or drum or a special plate. The stop spool or drum of the turret is linked kinematically to the turret by gearing in such a manner that when the turret is indexed to the next station, the stop spool or drum is likewise indexed to the next stop.

The component carrying the adjustable stops may be of various shapes, for instance, in the form of a shaft with longitudinal slots for clamping the stops, a drum of large diameter arranged in line with the turret and also having longitudinal slots, etc.

The size-maintenance travel-limiting device of a slide, head, etc., may be connected with a built-in instrument which automatically checks the corresponding dimension of the workpiece either periodically or continuously. When the required size is reached the instrument automatically disengages the feed or stops the machine.

Such devices which automate operation are employed especially frequently in machine tools for finishing operations, instruments of greatly varied types being used in them.

Devices for Protecting Machine Tools Against Overloads

The dimensions of each critical component of a machine tool, required to ensure sufficient strength and rigidity, are determined in designing on the basis of the whole complex of maximum forces P and torques M_t acting on this component. Hence, a device protecting this component against dangerous overloads should automatically restrict the force and torque to the limiting values P_{lim} and $M_{t\ lim}$. It is evident from the equation

$$N = C_1 P v = C_2 M_t n$$

where N = power transmitted
v = velocity
n = speed in rpm of the component being protected against overloads
C_1 and C_2 = constants,

that a limiting of the power in the corresponding kinematic train is equivalent, from the standpoint of efficient protection, to a restriction of the force or torque only at v = const or n = const, respectively, i.e., in single-purpose machine tools operating with invariable cutting conditions and only if the main and feed drives are powered by separate electric motors. If these trains are not powered by separate motors or if $v \neq$ const and $n \neq$ const, the restriction of the power of the motor to some constant limiting value N_{lim} will not maintain the values $M_{t\ lim}$ or P_{lim} constant. Therefore, devices limiting the power of the drive motor of the machine tool cannot, in general, replace such overload protection devices as shear pins,

clutches, etc., and do not eliminate the need for such devices as the latter. The two types of devices have principally different functions.

Electric, hydraulic and mechanical protection devices are extensively employed in up-to-date machine tools, several devices of different types often being used at the same time in a single machine. The most advanced type, in respect to their operation, are electric protection devices (though not applicable in all cases) and instantaneous-action safety trip clutches. The choice of a system of protection devices depends on whether their main purpose is to protect the machine, cutting tool or drive motor, on their required automaticity, on the rapidity of their operation and sensitivity.

Only mechanical overload protection devices are considered below. The following have found most widespread application in machine tools: (a) shear or breaking pins and keys; (b) friction, jaw (ratchet-tooth), ball-type clutches, etc.; and (c) dropping worms.

A flat belt also protects a machine tool against overloads; it will continue to slip until the load is reduced to the rated value.

It is insufficient in some cases just to stop the machine upon an overload; it is also necessary to reverse the motion as, otherwise, the cutting tool or machine may be broken when work is renewed (deep-hole drilling machines, etc.). In these cases, the protection device should be combined with a reversing mechanism.

Shear or breaking pins or keys. These components, installed in a suitable place in the kinematic train and joining together two shafts, or a shaft with a gear, sprocket, ratchet wheel, etc., are designed so that at $M_t > M_{t\ lim}$, the pin or key is sheared through or broken. This disengages the corresponding kinematic train and thereby avoids damage to more critical components of the machine or destruction of the tool.

Typical constructions of the shear pin are illustrated in Figs. 177 (Machine Tool Industry Std P95-10) and 178. As can be seen, the shear pin is inserted in most cases in hardened steel bushings, press-fitted into holes in the joined components. Therefore, the edges of the hole are not distorted when the pin is sheared. Part *a* keeps the shear pin from falling out.

The magnitude of the force required to shear the pin depends chiefly on the material of the pin, its heat treatment and its minimum diameter. Therefore, this force can be varied in a sufficiently wide range without changing the diameter of the hole for the shear pin by varying the pin material and its heat treatment, and by using pins with rectangular or vee-shaped necks of various diameters or plain pins in bushings with constant outside diameters and various inside diameters.

Shear pins are made of steel 45 (Machine Tool Industry Std P95-10), spring steels and structural steels 15, 20, 35, etc. The shearing bushings are most often made of steel 40X which is hardened and tempered to 48-53R_C.

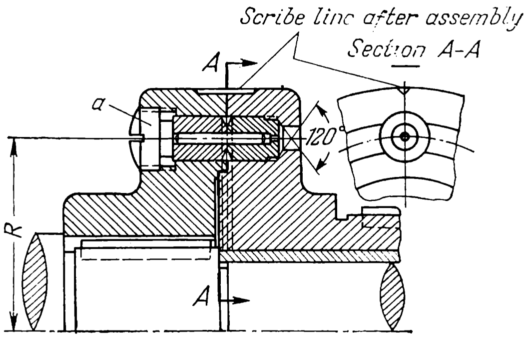


Fig. 177. Axial shear pin

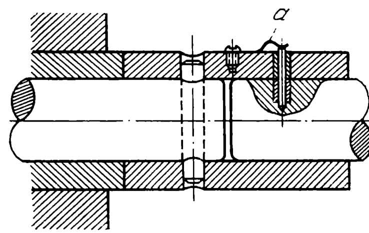


Fig. 178. Radial shear pin

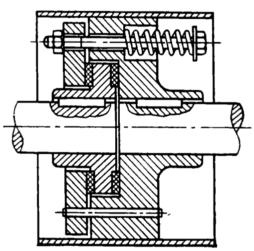


Fig. 179. Safety friction clutch

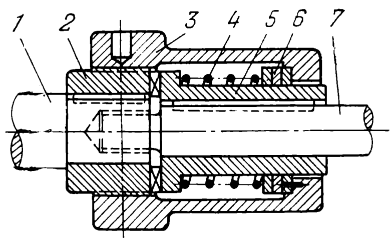


Fig. 180. Safety jaw clutch

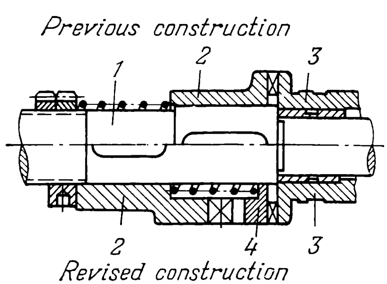


Fig. 181. Safety jaw clutch (two versions)

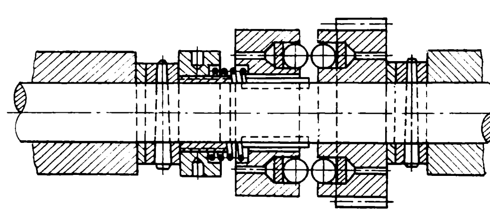


Fig. 182. Ball-type safety clutch

Shear keys are employed in a similar way in machine tools. They are made of the same material as shear pins, as well as of plastics.

Shear pins and keys are suitable in units in which overloads are infrequent.

Safety clutches. Safety clutches have the advantage over shear pins and keys that they are not destroyed by overloads but slip so that they disengage the corresponding kinematic train which they automatically restore as soon as the load drops to the normal value.

Safety (slip) clutches require only periodic adjustment or replacement of the worn parts.

Basically, any clutch can be used as a safety clutch if it is capable of self-disengagement when the transmitted torque exceeds a certain maximum value for the given clutch. Friction clutches are most commonly used for this purpose in machine tools. Safety friction clutches are similar in construction to ordinary friction clutches, differing chiefly in that they have no control components (Fig. 179). The service life of a safety clutch can be increased and the overloaded unit can be rapidly stopped by linking the clutch to some device for switching off the power.

The same materials are used for making the components of safety friction clutches as for those of friction clutches intended for any other purpose.

Jaw (ratchet tooth) clutches are also frequently used as safety devices and operate successfully if the angle of inclination of the sides of the jaws and the tension of the spring have been properly selected.

The clutch shown in Fig. 180 consists of sleeves (clutch members) *2* and *5* with jaws (teeth) having inclined sides and keyed on shafts *1* and *7*, respectively. In transmitting torques which do not exceed the established limit $M_{t\ lim}$ engagement of the clutch jaws is effected by the action of coil spring *4*. The pressure exerted by the spring, along with the value $M_{t\ lim}$, is varied by adjusting sleeve *3* axially. Thrust bearing *6* is needed here because spring *4* bears on one end against clutch member *5* which is rigidly linked to shaft *7* and on the other end against sleeve *3* which is linked through clutch member *2* to shaft *1*.

Disengagement of the jaws in safety devices of this type requires axial movement of one of the members with jaws. Experience shows that the friction resistance between this member and its guiding key or splines may sometimes be so large that no axial motion takes place and the clutch is not tripped. In Fig. 181, the upper half shows the safety clutch of the table feed mechanism of a planer-type milling machine while the lower half shows how excessive friction can be eliminated.

The clutch should link hub *3* of a gear (not shown) with the feed screw *1* of the table. The usual solution was applied in the earlier construction: hub *3* is freely mounted on a plain part of feed screw *1*; clutch member *2* is keyed to the screw. In the new design, shown below the centre line, inter-

mediate clutch member *4* with jaws having inclined sides has been inserted between hub *3* and clutch member *2* which is keyed on the feed screw. The jaws of member *4* engage identical jaws on hub *3*. Clutch members *2* and *4* are engaged together by five large square jaws with ground sides. In case of an overload, clutch member *4* slides easily to the left since its motion is not restricted by heavy friction on a key.

A ball-type safety clutch, shown in Fig. 182, operates on the same principle as a jaw clutch from which it differs only in that balls have been used in place of jaws. The balls are retained by riveting over the edges of the sockets into which they are inserted.

Design of Overload Protection Devices

The diameter of a shear pin is determined from the equations (for single- and double-shear pins, respectively)

$$\frac{\pi d^2}{4} = \frac{M_{t\,lim}}{R\tau_s} \quad \text{and} \quad \frac{\pi d^2}{4} = \frac{M_{t\,lim}}{2R\tau_s} \tag{218}$$

where d = pin diameter at the cross section where shearing is to occur
$M_{t\,lim}$ = design torque at which the pin is to shear; taken with a certain margin (about 20 to 25 per cent) in respect to the normal maximum M_t
τ_s = shear strength
R = distance from the shaft axis to the pin axis.

Substituting $\tau_s = k\sigma_t$, where σ_t is the tensile strength, in equation (218), we obtain

$$d = a\sqrt{\frac{M_{t\,lim}}{R\sigma_t}} \tag{219}$$

where $a = \sqrt{\frac{4}{k\pi}} = \frac{1.13}{\sqrt{k}}$ for a single-shear pin and $a = \sqrt{\frac{2}{k\pi}} = \frac{0.8}{\sqrt{k}}$ for a double-shear pin. According to experimental data $a = 1.20$ to 1.35 is suitable for single-shear pins and $a = 0.85$ to 0.95 for double-shear pins.

Machine Tool Industry Std P95-10 stipulates a pin size range of $d = 1.5$ to 10 mm for pins of steel 45.

The design torque $M_{t\,lim}$ depends not only on the operating conditions of the machine tool, but on where the shear pin (or other protection device) is located. It is essential that: (1) the driven part of the kinematic train stops as soon as possible after the pin is sheared (or device is tripped) and (2) that the protection device is not tripped or otherwise operated when the machine tool is started. It is also of importance that the safety device be

accessible for the replacement of the sheared or worn parts with spare ones or for adjustment of a clutch.

The kinetic energy of the part of the kinematic train that is to stop when the protection device is tripped should be as low as possible to ensure that the machine stops rapidly. Therefore, this device should be located so that no considerable flywheel masses are between it and the place where the force is applied that can cause an overload. It can be shown by analysis that in protecting against a dangerous overload by restricting the transmitted torque, the protection device should be arranged in the corresponding kinematic train so that the transmission ratio is constant in the part of the train between the device and the point where the force is applied that can lead to overloading.

To avoid tripping (or shearing) of the protection device in starting the machine tool (impermissible for shear pins and keys, and highly undesirable for clutches), it is necessary to comply with the condition

$$M_{t\,str} < M_{t\,lim} \tag{220}$$

where $M_{t\,str}$ is the maximum torque on the shaft on which the protection device is mounted during the period in which the machine is started.

To comply with this condition, it may sometimes prove necessary to locate the protection device on a shaft more remote from the final member of the kinematic train that is being protected against overload.

Shear keys are designed by employing the same calculations as for shear pins. Friction and jaw safety clutches are designed similarly to clutches of the same types intended for other purposes. The calculations include the coefficients of friction which can be estimated only approximately. Therefore, the construction of a safety clutch should allow it to be quickly adjusted in a sufficiently wide range.

CHAPTER 12

DYNAMIC CALCULATIONS AND ANALYSIS IN MACHINE TOOL DESIGN

12-1. Dynamic Performance of Machine Tool Systems

The higher requirements that are being made to the accuracy of the dimensions and geometric features of machined workpieces, the development of new materials that are difficult to machine, and the extensive introduction of process automation, leading to the design of machine tools with automatic systems of control and feedback adjustment, have greatly heightened the role of dynamic processes in machine tool design.

In the design, manufacture and operation of machine tools, engineers are more and more frequently confronted with problems concerning the dynamical effects.

These problems can be reduced to three main types:

(1) selection of the parameters of a drive;

(2) analysis of machine tool behaviour upon travel of the units without cutting action (idle-run operation of the machine tool);

(3) analysis of machine tool behaviour in the process of machining a workpiece (machine tool operation under load).

Along with experimental appraisal of pilot models, dynamic calculations in designing a new machine tool have acquired especial importance. The aim of such design and experimental appraisal is to compare several existing models or an existing model with one being designed, as well as various versions, in respect to their dynamic performance.

In addition to an appraisal of the machine tool, dynamic calculations cover a comparative appraisal and aid in selecting the construction of the cutting tool, fixture (clamping devices, etc.), cutting speeds and feeds, and the drive.

The indices of the dynamic performance of a machine tool are determined by calculation and experimental investigation on the basis of the general theoretical propositions presented below.

Indices of dynamic performance of machine tool systems include:

1. *Margin or degree of stability*. The loss of stability of a system is manifested by the occurrence of vibration or gouging of the cutting tool, by the nonuniform stick-slip travel of the units and their jamming. It becomes necessary to stop work on the machine tool and to attempt to eliminate the causes of these phenomena. The margin of stability defines the possibility

of changing some parameter of a system without the loss of its stability. We can, for example, speak of the margin of stability in respect to the rigidity of a boring bar or to its overhang, or in respect to the depth of cut, etc. It is convenient to express the margin of stability of parameters concerned with the frequency response of a system in the form of the margin of stability in respect to the amplitude or phase of this characteristic. The degree of stability is defined by the rapidity of decay of a process initiated in a stable system by external actions. In the case of vibrational processes, a convenient index of the degree of stability of a system is the damping factor taken as the characteristic of damping in the theory of vibration.

2. *Deviations of the parameters of a system upon external action*: (a) static deviations; (b) stationary dynamic deviations (in particular, forced vibrations); (c) transient dynamic deviations; and (d) random deviations.

The selection of parameters by which the indices of a system, subject to external actions, are to be determined should be guided by the specific objectives of the calculations or analysis, i.e., by the type of problem and the kind of criteria to be employed to evaluate the indices. Such criteria are: machining accuracy, service life (durability) of the machine tool, fixtures and cutting tools, production capacity, and energy losses.

Of prime importance are dynamic calculations and analysis in respect to workpiece machining accuracy. In this case, the indices of dynamic performance of the systems subject to external actions take the following forms:

(a) static machining errors, this indice being determined in machining a blank with constant machining allowances and upon invariable external actions on the system;

(b) stationary dynamic machining errors, in particular waviness or lobed-form of the machined surface upon forced vibrations;

(c) transient dynamic machining errors occurring as a result of deformation and other deviations of the system during transient processes, such as when the cutting tool is fed into or runs out of the cut (single-point tool or tooth of a milling cutter or broach);

(d) random dynamic machining errors which are the result of the action on the system of external factors having a random or chance nature.

The parameter by means of which the accuracy of a system is calculated or analyzed is the displacement of the cutting tool and workpiece in a direction normal to the surface being machined.

3. *Speed of response of a system*. This index determines the duration of the given transient process and is usually expressed by the time required by the process. The speed-of-response index is evaluated by criteria of accuracy, service life (durability), production capacity and the magnitudes of the energy losses.

In the following, all indices of dynamic performance of machine tool systems are illustrated by examples concerned mainly with machining accuracy.

12-2. Dynamic System of a Machine Tool

The dynamic system of a machine tool constitutes the aggregate of the *elastic system*—the machine-fixture-tool-workpiece complex (MFTW complex)—and the *working processes* occurring in the movable joints or associations of the components of the elastic system (cutting, friction, electrodynamic, hydrodynamic and other processes).

In machine tool operation, deformation of the elastic system occurs under the action of cutting forces, friction forces, forces developed by the motor, etc. For the sake of clearness, all the factors which affect the elastic system can be represented in the form of a diagram shown in Fig. 183. The elastic system and each working process affecting it are shown in the diagram by rectangles. Force effects and the deformation they cause are represented by arrows. Such a diagram is valid for the case in which the deformation of the elastic system does not lead to variations in the magnitude of the force, its direction or the character of load application (in other words, when the force is not a function of the deformation), i.e., to variations in the co-ordinates or the laws of the variations (the first and second derivatives of the co-ordinates of the system with time).

In this case, the forces acting on the system are external in relation to the system and can be constant or variable with time. For example, if some force varies according to a harmonic function, the elastic system will begin to execute forced vibrations.

In many cases, however, deformation changes the relative positions of the components of an elastic system constituting a movable joint (or association) and thereby changes the conditions under which any working process proceeds. This leads to a change in the acting force itself. Let us consider several examples.

1. The elastic system is deformed by the cutting force. This deformation alters the relative positions of the workpiece and tool constituting the movable association in which the cutting process occurs. As a result, the chip thickness is changed and, with it, the cutting force.

This change in the force affects the magnitude of the deformation, etc. This can be readily observed in the example of a lathe.

2. The elastic system is deformed by the friction force. This deformation changes the relative positions of the ram and guides constituting the movable association in which the friction process proceeds. A change in the normal load (normal contact deformation of the friction surfaces) leads to a change in the friction force and, consequently, in the deformation it causes. The circle of interaction has closed back on itself again. This can be exemplified by a saddle traversed along ways by a screw. Upon misalignment (cocking) of the saddle due to offset of the resultant of the friction forces in respect

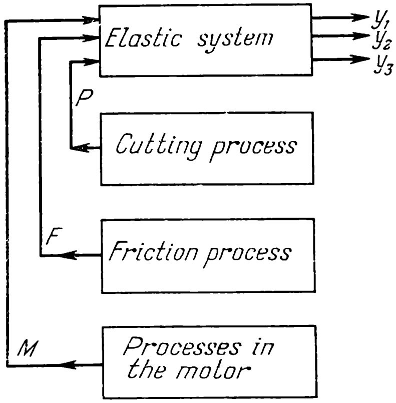


Fig. 183. Block-diagram of an open-circuit dynamic system of a machine tool

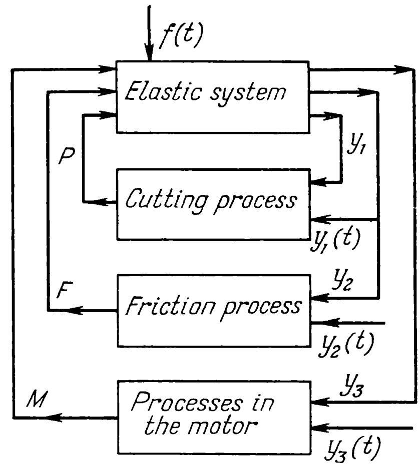


Fig. 184. Block-diagram of a closed-circuit dynamic system of a machine tool

to the screw axis, a friction force is developed on the side surfaces of the ways. This force varies with the variation in the deformation of the screw, its supports and drive.

3. The elastic system is deformed by the torque of the electric motor. This deformation changes the velocity of relative motion of the rotor and stator constituting the movable association in which electromagnetic processes occur.

In a motor with a drooping-speed characteristic, the motive force (torque) is consequently changed.

With this change in torque, the deformation of the elastic system, i.e., speed of motion, is changed, etc.

The forces acting on the elastic system in the examples given here cannot be considered external, since they vary with variations in the deformation of the system. The diagram in Fig. 183 showing the action of forces on the elastic system should be replaced by another diagram, shown in Fig. 184, in which the elastic system has an inverse effect on the cutting, friction and motor processes, etc., also shown by arrows. In necessary cases, the interaction of the elastic system with inertia forces, thermal effects, etc., should also be taken into consideration.

The remaining forces, not dependent upon the deformations of the elastic system, are external in reference to this system. Their effect on the elastic system is shown by the arrow f (t). In many cases, such forces include the

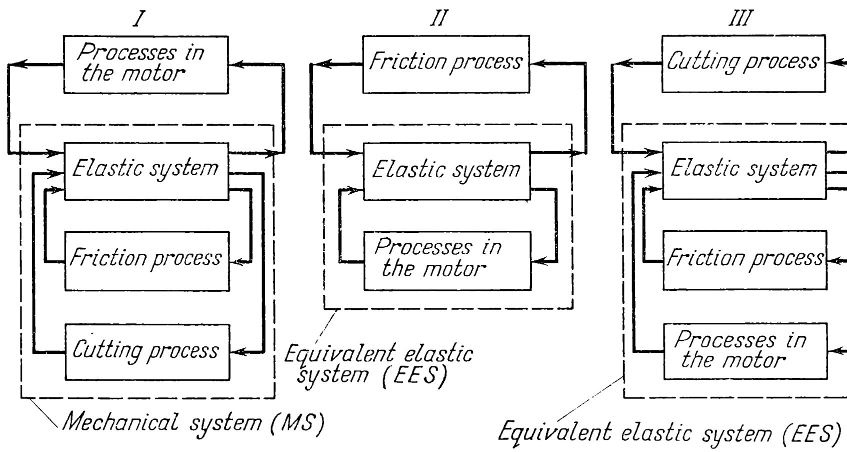


Fig. 185. Simplified block-diagrams of a closed-circuit dynamic system of a machine tool

inertia forces of unbalanced rotating components or of reciprocating units, forces due to the weight of the units and forces due to impacts and vibrations transmitted from outside through the foundation or originating in the system itself because of inaccurate meshing of gears or other errors in manufacture of the parts and their assembly, etc.

Changes in the chip cross section (actually the cross section of the undeformed chip), interaction of the friction surfaces or in the rotational speed of the rotor, etc., i.e., variations in the conditions under which the working processes proceed, may occur, not only as a result of deformations in the elastic system, but also from external causes (increase in machining allowance, variation in lubricant pressure, variations in voltage, etc.). These external effects acting on the working processes, which we shall call changes in adjustment or setting, are shown by arrows $y\ (t)$ in Fig. 184 referred to the corresponding elements of the dynamic system of the machine tool.

The elastic system and the working processes—cutting, friction and processes in motors—are the *principal elements of the machine tool dynamic system*. The interaction of the elements on each other are called the linkages; a train of such interaction is called the *linkage circuit*. A linkage circuit may be either of the open or closed type. An open linkage circuit is shown in Fig. 183 and a closed one in Fig. 184.

The dynamic system of a machine tool is a complex multiple closed circuit system. In application to the three types of problems mentioned in Sec. 12-1, it proves convenient to replace such systems with simplified one-circuit systems shown in Fig. 185. Taking advantage of the fact that the

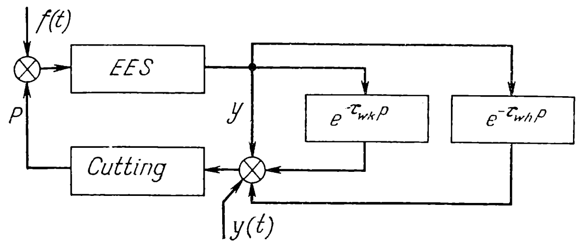


Fig. 186. Block-diagram of a machine tool dynamic system for machining "along chatter marks" in grinding

working processes (cutting, friction, etc.) interact only through the elastic system and have no other means of interaction, the elastic system together with a part of the working processes can be replaced by a certain element which is equivalent in respect to dynamic properties. This element may be either mechanical or another equivalent elastic system. The dynamic characteristic of such an equivalent element should be determined with account taken not only of the elastic system, but of the working processes included in the system and their interaction with the elastic system.

A feature of machine tool dynamic systems is the possibility of motion of the components of the elastic system "along chatter marks". Such, for example, is turning with a straight-turning tool, grinding with repeated passes, etc. In this case, the dynamic system is modified: a supplementary delayed feedback is added. This means, for example, that the deformation of the system, taking the form of waves on the surface of the workpiece being machined, is fed back into the system, after one revolution of the workpiece, as a change in the thickness of the layer of stock being removed. Shown in Fig. 186 is the diagram of a system with two delayed feedbacks: in respect to the grinding wheel which wears in the course of operation and in respect to the workpiece being ground.

In the following, cases of operation "without previous chatter marks" and "along chatter marks" will be considered separately.

It should be noted that each of the principal elements of a machine tool dynamic system comprises a complex system which is revealed in analysing the processes proceeding in these elements. The elements of a dynamic system may be either stable or instable. It is in this sense that the inherently instable elements of a system are mentioned further on.

We shall call a physical quantity, describing the action on the given element or system, the *input co-ordinate* of the element or system; the result of the action—the *output co-ordinate*. For instance, the input co-ordinate of

an elastic system is the force action while the output co-ordinate is the deformation it causes.

It is not by chance that the idea of the closed-circuit nature of machine tool dynamic systems has been treated here with such emphasis. It is due to the highly essential difference in the dynamic properties of closed- and open-circuit systems.

The main differences between closed- and open-circuit systems are:

1. An open-circuit system consisting of instable elements is instable; one consisting of stable elements is stable.

A closed-circuit system, on the other hand, consisting of stable elements may turn out to be instable, and conversely may turn out to be stable even when it contains instable elements.

2. A closed-circuit system reacts or responds entirely differently to external actions than open-circuit systems do.

Examples illustrating these propositions follow.

To provide convenience in their analysis, dynamic systems can be dismembered, "disconnecting" the linkages between the elements. If one of the linkages is disconnected, the system is called a *disconnected system*. If both linkages of an element are disengaged, the element can be singled out of the system and considered separately, investigating the relationship between the input and output co-ordinates.

The properties of an element of a dynamic system or of a chain of elements constituting a disconnected system are determined by the relationship between the input and output co-ordinates of the element or system. We shall call this relationship the *characteristic of the element* or *system*. If it is obtained under the conditions of a stationary process, when the input co-ordinate does not vary with time, then the characteristic will be *static*. The same relationship, obtained in the case of an action that varies with time, will be a *dynamic characteristic*.

Real characteristics of elements and systems are nonlinear as a rule. For example, the static characteristic of an elastic system, i.e., the relationship of the deformation of the elastic system of a machine tool to the acting forces is expressed by well-known loop-shaped curves. To simplify analysis, the characteristics are linearized, i.e., presented in the form of linear differential equations. If these equations are written in operational form, the dynamic characteristic is called the *transfer function* of the element or system.

Of convenience in calculations is the so-called frequency dynamic characteristic, determined under conditions of variable input co-ordinates with time according to the law of harmonic oscillation. The frequency of this oscillation varies theoretically from zero to infinity while practically it varies within the limits of the frequency range which may interest us and which is called the working range.

The relationship of the ratio of the amplitudes of oscillation of the output and input co-ordinates to the frequency gives the gain-frequency characteristic; the relationship of the phases of oscillation gives the phase-frequency characteristic. The combination of these two characteristics (in complex co-ordinates) is the *gain-phase frequency characteristic* (GPFC). There are also other types of frequency characteristics (real, imaginary, log, etc.).

The ratio of the output to the input co-ordinate in an elastic system, written in complex form, is called the *dynamic unit deflection* while the reciprocal of this ratio is called the *dynamic rigidity*.

The so-called time-response dynamic characteristics, obtained for a given law of variation of the input co-ordinate with time, are widely applied. In most cases, a stepped variation of the input co-ordinate, from one steady value to another, is taken. Both static and dynamic characteristics can be represented either graphically or analytically. Characteristics can be constructed theoretically and obtained experimentally. In plotting the characteristic from experimental data, the selected variation of the input co-ordinate is produced by a special device and the corresponding variation of the output co-ordinate is registered. For example, in determining the frequency characteristic of an elastic system, the input action, substituting for the force of cutting, friction or the motor, is produced by a vibrator, and the deformation, in the direction that we are interested in, is registered by some kind of displacement or velocity pickup.

Using amplitude and phase meters or recording the variations in force and displacement on an oscillogram, and then suitably processing these data, the frequency characteristic is plotted.

Static and dynamic characteristics of an elastic system and the working processes are given in the following.

The concept of a machine tool dynamic system accepted here differs from the one found in literature treating with problems of the deformability of a machine tool elastic system and its influence on the stability and accuracy of the machining process or on idle-run machine operation. The present concept provides clearness and convenience in analysis; results are obtained in a comparatively simple manner. The terminology, many ideas and the methods of analysing problems of machine tool dynamics have been adopted from automatic control theory whose problems, as our analysis of systems shows, are very close to the problems of machine tool dynamics. The fundamentals of control theory can be found in the corresponding literature.

The interaction between the elastic system and the working processes is accomplished, on the one hand, through the forces initiated by the working processes and, on the other hand, through the parameters of the processes affecting the variation in the forces and being changed upon deformation of the elastic system. These parameters are very numerous. There is also a great number of components of the resultant force of any working process.

Therefore, it becomes necessary to select the most essential of them for definite conditions of machine tool operation.

Experience shows that the variation in the cutting force is determined primarily by the variation in the cross-sectional area of the undeformed chip. In the usual case, the area of the chip cross section varies to a greater degree when the thickness of the chip is varied than when its width is varied. Thus, as a first approximation for large and medium undeformed chips, their thickness can be taken as the parameter determining the variation in the cutting force.

In case of small cross-sectional areas of the undeformed chip (light chips) and the presence of a worn area on the tool flank, the variation in cutting force is determined to a greater degree by the friction on the tool flank which depends upon the contact deformation of the workpiece surface in a direction coinciding with the direction of the undeformed chip thickness. Thus, the linkage being considered between the cutting process and the elastic system is determined, as a first approximation, by displacements perpendicular to the cutting surface.

Comparatively small variations in the undeformed chip thickness change the position of the resultant of the cutting forces only to a small extent in many cases.

It can therefore be assumed as a first approximation that the input co-ordinate of the elastic system in respect to the cutting process is the resultant cutting force, while the output co-ordinate is the deformation in the direction of the normal to the cutting surface. In considering the cutting process, the input and output co-ordinates are interchanged correspondingly.

The friction force, in the case of contact of rubbing surfaces, i.e., for the conditions of dry (in the conventional practical sense) or boundary friction, is determined by the variation in the normal load or, in other words, the normal contact deformation of the friction surfaces. In many cases, characterized by a slight dependence of the coefficient of contact friction on the load, the resultant of the friction forces varies in direction to only a small extent upon variations in normal contact deformation. Therefore, it can be assumed as a first approximation that the input co-ordinate of the elastic system in respect to the friction is the variation of the resultant friction force, while the output co-ordinate is the deformation in the direction of the normal to the friction surface. In considering the friction process, the input and output co-ordinates are correspondingly interchanged.

The force or torque developed by an electric or hydraulic motor is determined primarily by the velocity of relative motion of the rotor and stator or piston and cylinder. Hence, in many cases, the input co-ordinate of the elastic system in respect to the processes in motors is the force or torque developed by the motor, while the output co-ordinate is the variation in the velocity of relative motion of the components of the elastic system (rotor

and stator or piston and cylinder) which is determined by the deformation of the system. In considering the working processes in motors (electromagnetic, hydrodynamic, etc.), the input and output co-ordinates are correspondingly interchanged.

Aero- and hydrodynamic processes, as well as electromagnetic processes, can occur at other parts of a machine tool besides motors. They are found in sliding friction bearings, slideways and certain other special devices. In these cases, as for the more complex dynamic systems of machine tools, the input and output co-ordinates of the elements are determined on the basis of special analysis.

When the direction or point of application of the resultant force of some working process is changed, it proves convenient to go over to separate treatment of its normal and tangential components. The dependence of the normal component on the above-mentioned deformations of the system along the normal to the cutting or friction surface takes the form of the rigidity of the movable association of the elements of the elastic system. Thus we have "cutting rigidity", "friction rigidity", "lubricant film rigidity", etc.

12-3. The Elastic System

The elastic system includes the machine tool, fixture, cutting tool and the workpiece. The system has an infinitely large number of degrees of freedom and can only be considered approximately as a system with several degrees of freedom.

The inherent instability of the elastic system in machine tools is found, practically, in the following cases:

(a) in machining workpieces rotating at a speed near to the critical speed;

(b) in the operation of a long, thin centrally positioned tool (twist drill, core drill, etc.) or in machining long machine tool parts (screws, piston rods, etc.) under conditions of buckling;

(c) in machining thin-walled parts or when thin-walled parts are used in the machine tool.

In practical analysis and calculations, it becomes necessary to resort to the conception of equivalent elements of the elastic and mechanical systems. These systems include a great number of movable associations or joints of the components in the machine tool system. Displacement of the slides, tables, heads, etc., in the sense of travel required to accomplish the given processing operation, takes place along these joints. Friction and other working processes, as mentioned above, have a very great effect on the stability of the system and on its static and dynamic characteristics. Conditions are created for the initiation of instability which, in these cases,

is manifested as self-excited vibrations of the transmissions, bearings and like movable associations that were included in analysis in the equivalent elastic or mechanical system. These self-excited vibrations are usually coincident with the forced vibrations caused by errors in the manufacture and assembly of the parts (runout of pulleys, local thick places in belts, backlash in gearing, waviness in the races of antifriction bearings, etc.).

Similar effects are also observed in the supporting system upon the travel of slides, crossrails, heads, etc., in the machining process. In many cases, other processes in movable associations, such as processes in motors, play the same role as friction.

In designing and developing a machine tool, fixture or cutting tool in actual practice, efforts are always made to eliminate all kinds of instability of the elastic system by getting out of the zone of critical rotational speeds or buckling conditions, and by creating conditions for stable travel of all units and parts of the machine.

The characteristics of an elastic system are determined by the following principal parameters: masses or moments of inertia of the units and parts, rigidity of the elastic elements, forces of nonelastic resistance (damping), and links between the displacements of the masses in a system with many degrees of freedom.

The use of units and parts with large masses and moments of inertia leads to a reduction in the natural frequencies of the system and to an increase in inertia loads and the duration of transient processes. Changes in the mass and moments of inertia in machine tools are usually associated with changes in the elastic properties of the construction. Thus, for instance, a reduction of the mass of a column due to a reduction in the thickness of its walls or alterations in its configuration, inevitably leads to changes in the rigidity of the column. By selecting a rational shape and adding stiffening ribs, a certain reduction in mass can be achieved without reducing the rigidity.

Rigidity is defined as the ratio of the forces causing deformation to the magnitude of the latter. It is frequently more convenient to use the reciprocal of rigidity which we have called the unit deflection of the elastic system.

The deformability of a machine tool elastic system depends upon the rigidity of the components, the contact deformation in the joints between the components and the local deformations of the elements of the construction which serve to form the joint (various lugs, strips, feet, etc.).

Depending upon the size and configuration of the component (shaft, bed, column, headstock, etc.) and its joint with the mating component, the magnitude and relative significance of any one kind of deformation in the general deformation of the system may vary in a wide range. A proper assessment of this fact is important in choosing a method of increasing the rigidity of a construction.

The forces of nonelastic resistance or the damping forces in a machine tool are determined mainly by the friction in the joints or associations of the components. If the components forming a joint do not have a given relative motion in the case being considered, and relative displacement occurs only as a result of deformation of the system, the friction forces always damp the vibrations, dissipating a part of the energy introduced into the elastic system when it is loaded. If the given motion or travel exists, the damping action of friction is determined by the degree of stability of the corresponding system. In this case, as a rule, the friction forces increase the amplitude of forced vibrations of the system when it is subjected to external actions.

A damping effect is always manifested by the forces of "viscous" friction, i.e., forces proportional to the velocity. Such "viscous" friction may be evident, not only in a viscous medium, for example, a lubricant, but in joints with dry friction.

The large and, at the same time, opposing in its effect role played by the joints and associations of the machine tool components in damping the vibrations of the elastic system and in its deformability, substantially complicates the problem of whether it is expedient to eliminate or introduce a certain joint. The introduction of a new joint lowers the rigidity of the construction in comparison with an integral construction (one without the given joint) but, at the same time, increases the damping effect. This may turn out to be more essential from the point of view of eliminating vibration in machining and obtaining the required surface finish on the workpiece. This explains cases, sometimes encountered, in which vibration in cutting is eliminated by loosening certain joints between the units of a machine tool. For example, in milling a workpiece in a heavy planer-type milling machine, vibrations appeared even though all joints were drawn up tight in order to increase the rigidity of the system. Thus clamped, in particular, was the joint formed by the milling head and the housing ways along which no units travelled during the milling process. Vibrations ceased after unclamping the head on the ways.

The most important feature of an elastic system, ensuing from the fact that it has so many degrees of freedom, is the interdependence or interrelation between the deformation of the various elements or between the kinds of these deformations.

This interrelation is manifested by the fact that an attempt to produce a certain deformation by means of a given force will lead, as a rule, to another or other deformations. Thus, for instance, the bending of the workpiece by the action of the cutting force is necessarily accompanied by its twisting, since the direction in which the cutting force acts does not pass through the axis or centre of the workpiece.

Elastic, velocity and inertia links are found in the elastic system of a machine tool. The features of these links are treated in detail in a study course on vibration theory.

In assessing the influence of the links between the deformations of an elastic system on the dynamic processes, a very important factor is the nearness of the frequencies of the interacting vibrational systems, called partial systems, corresponding to each of the linked deformations. If the frequencies of free vibrations of the partial systems are near to each other, then even with a weak link between them the interaction of these systems turns out to be very strong.

A special term "internal resonance" can be found in literature on this subject. It defines the degree of coincidence of the frequencies of two linked partial vibrational systems. For example, at a definite diameter-to-length ratio of a boring bar, internal resonance may be set up between its flexural and torsional vibrations.

The characteristics of an elastic system are determined as the ratio of the displacements, representing reverse action of the deformation on the working process, to the external forces substituting for the forces of the working processes. For instance, the characteristic of the elastic system in respect to the cutting process is determined as the ratio of the displacement, normal to the cutting surface, to a force which imitates the cutting force. In respect to friction, the characteristic is determined as the ratio of the displacement, normal to the friction surface, to the force which imitates the full reaction of friction (or, in taking "friction rigidity" into account, to its tangential component). In analysing processes concerned with stopping in a movable association (relaxational self-excited vibration, etc.), the ratio of the tangential displacement to the tangential component of the cutting or friction forces is also determined.

Practically, in machine tool analysis or calculations, the characteristic of the equivalent elastic system (EES) or of the mechanical system (MS) is investigated.

As an example, we shall consider the characteristic of an equivalent elastic system in respect to the cutting process (a problem of the third type) in somewhat more detail. The dynamic characteristic of the EES is determined in this case upon variation of the external force, imitating the cutting force, with time in accordance with some law. The dynamic characteristic (dynamic unit deflection or its reciprocal—rigidity) of the EES is determined either by calculations or experimentally.

Methods of calculating the dynamic characteristic are being developed more and more at the present time in connection with the necessity for assessing the stability and other characteristics of dynamic performance of a machine tool in the design stage. The calculation of the characteristic in the frequency form consists in calculating the forced vibrations of the

EES under the action of a force imitating the cutting force and varying according to a harmonic law.

Difficulties in these calculations are associated with the complexity of the multiple-mass system of a machine tool, the necessity for taking into account the action of the working processes included in the EES (friction, hydrodynamic, etc.) and the complications in determining the parameters of the system (rigidities, damping, etc.).

At the present time, the calculation of the dynamic characteristic of the EES has become possible due to the availability of electronic computers in research and design institutions.

Calculations are carried out along the lines of present-day vibration theory, taking into account the interrelation of the partial vibrational systems, damping in the fixed and movable joints and associations of the machine tool components, etc. The choice of the number of degrees of freedom to be taken into consideration is determined by the complexity of the system and by the working range of frequencies. The generalized forces along the corresponding co-ordinates represent the action of the working processes on the elastic system. These forces, except for the one accepted as the input co-ordinate, using the dynamic characteristic of the corresponding processes, can be represented in the form of equations, expressing the dependence of the generalized forces on the generalized co-ordinates and their derivatives. The coefficients of these equations are added to the coefficients of the equations for the elastic system. As a result, we obtain a system of equations which, in form, contains only a single generalized force. This generalized force is either the input co-ordinate of the EES or one of the external actions on this system that we are interested in (in this case, the input action is equated to zero). The system of equations can be written as

$$\left.\begin{array}{c}
a_{11}\ddot{q}_1 + a_{12}\ddot{q}_2 + \ldots + a_{1n}\ddot{q}_n + b_{11}\dot{q}_1 + b_{12}\dot{q}_2 + \ldots + b_{1n}\dot{q}_n + \\
+ c_{11}q_1 + c_{12}q_2 + \ldots + c_{1n}q_n = k_1 Q_1 \\
a_{21}\ddot{q}_1 + a_{22}\ddot{q}_2 + \ldots + a_{2n}\ddot{q}_n + b_{21}\dot{q}_1 + b_{22}\dot{q}_2 + \ldots + b_{2n}\dot{q}_n + \\
+ c_{21}q_1 + c_{22}q_2 + \ldots + c_{2n}q_n = k_2 Q_1 \\
\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \\
a_{n1}\ddot{q}_1 + a_{n2}\ddot{q}_2 + \ldots + a_{nn}\ddot{q}_n + b_{n1}\dot{q}_1 + b_{n2}\dot{q}_2 + \ldots + b_{nn}\dot{q}_n + \\
+ c_{n1}q_1 + c_{n2}q_2 + \ldots + c_{nn}q_n = 0
\end{array}\right\} \qquad (221)$$

where $q_1, q_2 \ldots q_n$ = generalized co-ordinates of the system

Q = input (or external) action on the equivalent elastic system

k_1 and k_2 = coefficients for the components of the action along the different co-ordinates

a_{ij}, b_{ij} and c_{ij} = constants of the equations.

If $Q = Q_0 \sin \omega t$, in accordance with the definition of the frequency dynamic characteristic, then it can be calculated by the well-known method used to determine the forced vibrations of a system subjected to the action of a given external force.

The solution of the system of equations (221) can be written as

$$Dq_i = Q_1 \sum_{i=1}^{n} \sum_{j=1}^{n} M_{ij} \tag{222}$$

from which the transfer function of the equivalent elastic system is

$$W_{EES} = \frac{q_{out}}{Q_{in}} = \frac{\sum_{1}^{n} M_{ij}}{D} \tag{223}$$

where D = principal determinant of the system of equations
M = corresponding algebraic complement to determinant D.

As an example of the calculation of the dynamic characteristic of the equivalent elastic system in respect to the cutting process, we shall consider the design of a shaper. The design diagram is shown in Fig. 187.

On the basis of available experimental data on the modes of vibration of the shaper in the working range of frequencies, the system is represented as one with eight degrees of freedom. We shall examine a shaper doing grooving operations. In this case, the elastic system can be assumed to be plane. Since the feed motion is accomplished between working strokes, the joints and associations of the table, rail and column ways, as well as the other associations of the feed mechanisms are fixed. Analysis shows the absence, as a first approximation, of significant relationship between the longitudinal and transverse displacements of the ram. Therefore, longitudinal displacements of the ram, i.e., displacements along the Z axis can be disregarded in calculating the output co-ordinate. This enables the analysis of the movable association between the ram and its guides and that of movable associations in the drive train of the shaper to be excluded since these are closed-circuit systems.

Thus we obtain the simplest analog of the elastic system of the shaper. The absence of any significant effect of column deformation, as has been experimentally established, enables the elastic systems of the ram and table with the rail to be considered as independent systems whose deformations are added together. The system of equations for ram motion is of the fourth order; the system for the table with the rail is of the twelfth order. Fig. 188 shows the frequency dynamic characteristics of these two systems and the total gain-phase characteristic of the equivalent elastic system of the shaper. The calculations were carried out in ENIMS. Details concerning the calcu-

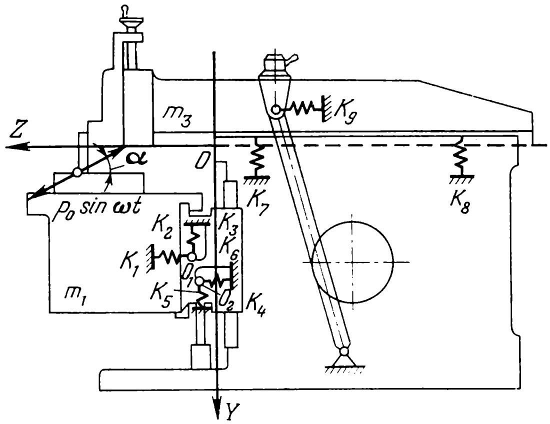


Fig. 187. Design diagram of the equivalent elastic system of a shaper

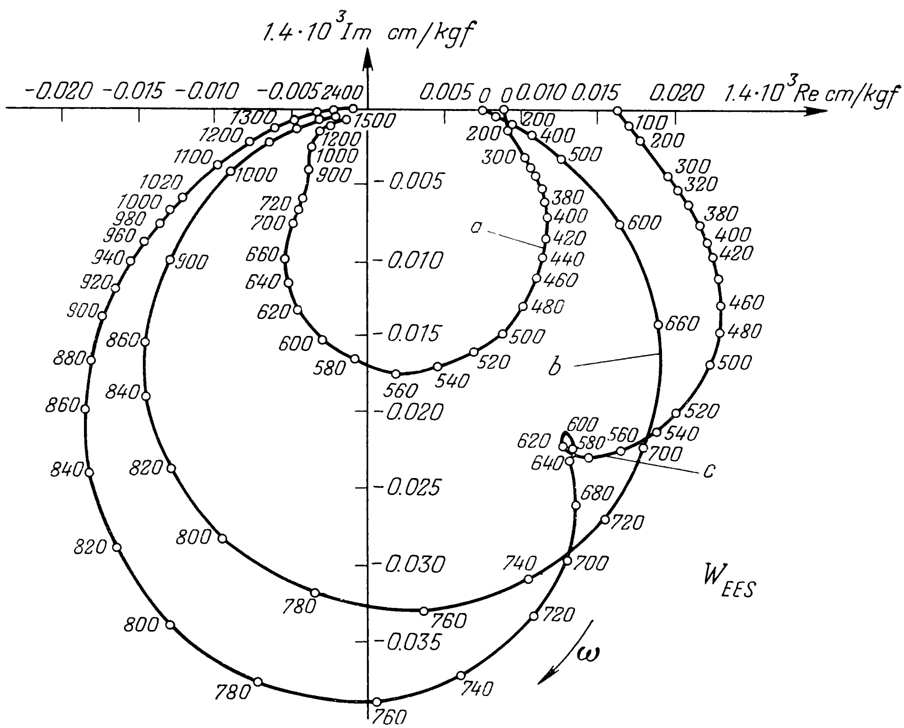


Fig. 188. Gain-phase frequency characteristics of the EES of the model 7B36 shaper: *a*—table system; *b*—ram system; *c*—total; *Im*—imaginary axis; *Re*—real axis; ω—circular frequency, 1/sec.

lation of the parameters of the system and the frequency characteristic are dealt with in the works of V. Kudinov and B. Nikitin.

In many cases in analysing complex equivalent elastic systems of machine tools, it proves convenient to change from randomly selected co-ordinates to the so-called normal co-ordinates. This operation, i.e., the conversion of the motion equations of the system, is called normalization.

The displacement that is of interest to us is obtained by adding together the displacements along the various normal co-ordinates. Independent equivalent elastic systems, having equations in normal co-ordinates, can be represented as separate elements of the system. The equations of these systems are of the second order. As a rule, they are vibrational systems. Their characteristic (transfer function) takes the form

$$W_{EES}=\frac{y}{P}=\frac{K_{EES}}{T_1^2p^2+T_2p+1} \tag{224}$$

where K_{EES} = reduced static characteristic (unit deflection) of the given normal form

$T_1=\frac{1}{\omega_n}$ = inertia time constant of the given normal form (reciprocal of the natural circular frequency ω_n)

$T_2=\gamma T_1$ = damping time constant of the normal form

$\gamma\cong\frac{\lambda}{\pi}$ = damping ratio

λ = logarithmic damping decrement

$p=\frac{d}{dt}$ = differentiation symbol

P = external force (input co-ordinate)

y = deformation (output co-ordinate).

As a whole, the equivalent elastic system can be represented as a system of elements connected in parallel in which the input co-ordinate is the same force and the output co-ordinates are added together algebraically. A diagram of this system is shown in Fig. 189. Of especial interest is the case in which the characteristic of at least one normal form has the minus sign. If in summing up the gain-phase frequency characteristics with a single sign, the full characteristic of the EES cannot cross the negative real axis (the characteristics are considered to be positive), then in this case the full characteristic covers four or more quadrants and crosses the negative real axis. Fig. 189 shows the summing up of the gain-phase and real frequency characteristics of two normal forms for both cases.

The attempt to simplify calculations justifies the change to the simplest type of vibrational system with one or two degrees of freedom. Such a conversion is possible in cases when the characteristic of the EES, in the frequency range that interests us, can be approximated by one or the sum of two

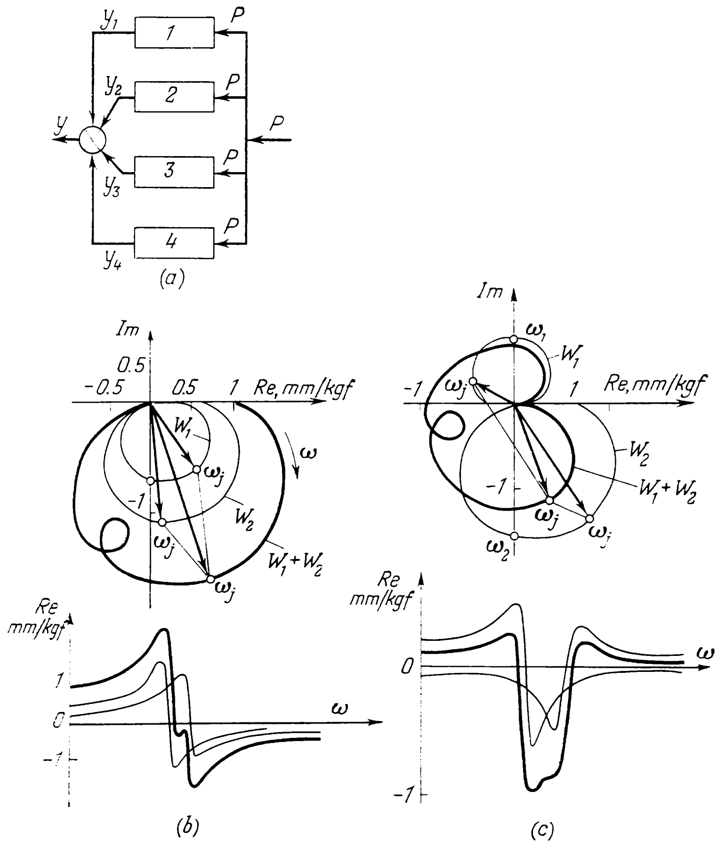


Fig. 189. Block-diagram and gain-phase frequency characteristics of the EES in normal co-ordinates:

(a) block-diagram of the system; (b) summing up characteristics with the same sign; (c) summing up characteristics having different signs

characteristics of the second order, i.e., characteristics of the corresponding normal forms.

As mentioned above, a force varying according to a harmonic function is produced by a vibrator when the frequency dynamic characteristics are determined experimentally.

A simple method is used to obtain stepped variation of the force. It consists in rapidly removing the force, imparted by a weight hanging on a wire or strong cord, by severing the wire or cord.

A pulse load is produced at the present time by a sufficiently primitive method—a hammer blow or a falling weight.

The readings of the force and displacement pickups are either recorded by means of an oscillograph or are noted down from the readings of special instruments.

A determination of the mode of vibration turns out to be very useful in analysing the roles played by the various units and parts of the machine tool, the workpiece and the cutting tool in the vibrations of the system. For instance, the modes of vibration for two natural frequencies within the working range have been investigated for radial drills. Here the lower frequency corresponds to the vibrations of the frame formed by the units of the radial drill, vibrating like a tuning fork due to the flexural deformation of the column and arm. The higher frequency corresponds to twisting of the arm and swivelling of the drill head.

Another method of revealing the roles played by the various parts and units in the vibrations at a certain natural frequency is the method of analysing the variation in the natural frequencies of the system upon varying the parameters of different parts of the machine tool (mass, rigidity, etc.).

Both described methods of experimentally assessing the roles played by the elements of the elastic system in the vibrations are only approximate but can be useful in the practical solution of problems associated with machine tool vibrations.

The information obtained by the aid of these two methods substantially supplements the data on the natural frequencies of vibration of the elastic system which are determined first of all in practice.

Assuming in the equation of the dynamic characteristic that $p = 0$ or $\omega = 0$, we obtain the static characteristic of the EES. The acting force is applied statically in the experimental determination of the characteristic. In respect to the cutting process, the acting force is the one imitating the cutting force.

In literature on the subject, as well as among engineers and scientists engaged in research concerning the deformability and accuracy of machine tools, a different characteristic of the elastic system in respect to the cutting process has been extensively employed. This is the "rigidity" of the machine tool or the machine-fixture-workpiece-tool (MFWT) system. The most popular and widely accepted definition of this term is the one proposed by A. Sokolovsky. He defined the rigidity of the system as the ratio of the component P_y of the cutting force (in reference to lathe operation) to the displacement along the Y axis (in the generally accepted system of co-ordinates), determined for the action of the full cutting force. In some cases, a reciprocal of this characteristic, the so-called unit deflection, is used. If the rigidity is measured in kgf per mm, unit deflection is measured in mm per kgf.

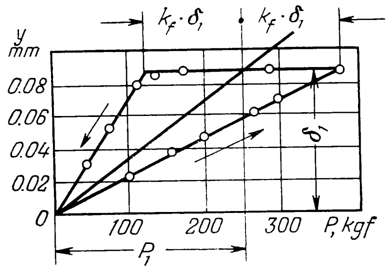


Fig. 190. Displacement curve for the elastic system of a lathe subject to static loading

It is not difficult to see what is common to the definitions of the static characteristic of an elastic system and the unit deflection or the rigidity. It is more expedient to use the concept of a "static characteristic", and not the "rigidity" or "unit deflection", employing the last two terms to characterize only the elastic properties of the construction. However, taking into account the popularity of the term "rigidity", we shall use it in the necessary cases (as defined by A. Sokolovsky), calling it the "processing rigidity" of the system or, correspondingly, the "processing unit deflection".

The static characteristic (processing unit deflection) of an elastic system is determined by the calculation, experimental and combined (experimental-calculation) methods. The calculation method has been mentioned above.

Experimental methods of determining the static characteristic (processing rigidity) have found wide application. These methods can be divided into two groups depending upon the method of loading the elastic system: (1) static loading by forces imitating the cutting force, and (2) by loading with the cutting forces in the process of machining a workpiece. The methods of the first group are called static methods; those of the second group are called production methods.

The load imitating the cutting force is applied by various devices (screw or hydraulic jacks, weights, etc.).

The deformation is measured by universal measuring facilities of the required accuracy (dial indicators reading to 0.01 or 0.001 mm, etc.).

A great number of instruments have been developed for static measurement of the processing rigidity. These include instruments of well-known design used in standard tests of various types of machine tools. The values of the force and the corresponding displacements obtained in the test are plotted graphically as shown in Fig. 190. The loop shape of this curve (non-linear characteristic) is due to the action of the friction forces in the joints. The increase in the width of the loop with the increase in deformation characterizes the proportionality of the friction forces to the magnitude of the deformation.

When the load is applied, the displacement of the system is less than it would be if there were no friction forces since a part of the acting force is used to overcome the friction forces. The opposite will be observed in unloading because the elastic force, in moving the system in the reverse direction, must overcome, not only the applied force, but the friction force as well since the direction of the latter now opposes the elastic force. To evaluate the elastic component of the static characteristic of the system, it is necessary to draw a line through points bisecting the intercepts between the loading and unloading branches of the curve. This line determines the actual rigidity of the system ($C_1 = \frac{P_1}{\delta_1}$ kgf per mm) as a characteristic of its elastic properties. The shape and area of the loop in the curve characterize the friction forces in the system. The intercept $2k_f\delta_1$ is equal to twice the friction force at the deformation δ_1.

The experimental determination of the static characteristic of an elastic system by the so-called production methods is widely used at the present time. The essence of these methods consists in evaluating the errors in machining a workpiece caused by deformations of the elastic system. The cutting conditions (speed, feed and depth of cut) and the shape of the workpiece are strictly stipulated. Knowing the machining error and having determined the cutting force either experimentally or by calculations, the processing rigidity or unit deflection of the elastic system can be computed. In other words, knowing the static error of a closed-circuit dynamic system and the static characteristic of one of its elements—the cutting process—it is possible to determine the static characteristic of the second element—the elastic system.

Properly applied, production methods can yield data on the static characteristic of the elastic system as valuable as the data obtained by the above-described static methods.

In respect to processing rigidity determined by production methods, literature on the subject sometimes uses the term "dynamic" rigidity of the machine tool. This term may lead to confusion. In vibration theory, the dynamic rigidity is defined as the ratio of the force to the displacement in the vibration of systems with a random frequency. In essence, this corresponds to the dynamic characteristic of an elastic system in the frequency form.

Using certain concepts, a convenient calculation method can be applied in many cases for determining the static characteristic by means of a simplified analog of the elastic system found by experiment. We have called this the combined (experimental-calculation) method.

The concepts on which this method is based can be explained by the example of the elastic system of a lathe, considering separately displacement of the cutting tool upon deformation of the carriage unit and displacement of the workpiece upon deformation of the spindle unit.

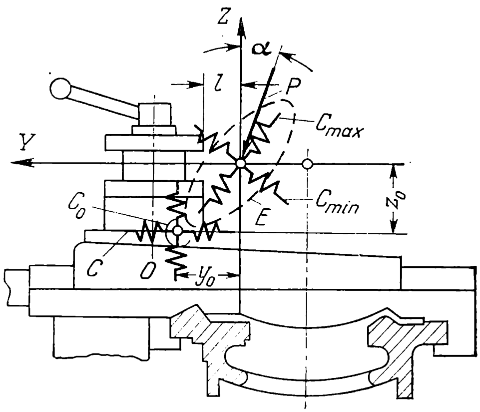


Fig. 191. Diagram of the elastic system of a lathe carriage unit:
O—centre of rigidity; E—ellipse of rigidity

In a plane perpendicular to the line of centres of the lathe, the elastic system of the carriage unit can be represented, for simplicity, by a system (Fig. 191) having a centre of rigidity. It will be recalled that in *Strength of Materials* the centre of rigidity or centre of twist is the point in reference to which the moment of internal elastic forces is equal to zero.

If a force passes through the centre of rigidity, then the displacement of any point on the compound slide (square turret) is determined by deformations along two principal central axes of rigidity without swivel of the square turret. If the force does not pass through the centre of rigidity, displacement from swivel about the centre of rigidity is added to the previously mentioned displacements. The magnitude of the displacement depends upon the moment of force P and the torsional rigidity C_0.

Since the displacement of the tool nose due to swivel of the square turret is large in comparison with the displacement of the centre of rigidity, the above-described system can be replaced by a simpler one, referred to the tool nose and determined by the two rigidities C_{max} and C_{min}. The axis of maximum rigidity is directed toward the centre of rigidity; the axis of minimum rigidity is perpendicular to the first axis. The value of the minimum rigidity is dependent upon the rigidity C_0 and the distance from the tool nose to the centre of rigidity.

Table 4 lists the values of the co-ordinates of the centre of rigidity, the torsional rigidity and the maximum and minimum rigidities referred to the tool nose of the carriages of lathes with a height of centres equal to 200 mm.

The ellipse of rigidity shown in Fig. 191 is used to determine the displacement of the tool nose from force P by calculations or graphically. It is

TABLE 4

Characteristic for a tool overhang $l = 35$ mm		Lathe No. 1	Lathe No. 2	Lathe No. 3
Co-ordinates of the centre of rigidity, mm:				
	Y_0	53	111	107
	Z_0	110	152	148
Torsional rigidity C_0, kgf per mm		76×10^6	130×10^6	81×10^6
Maximum rigidity C_{max}, kgf per mm		8.9×10^3	7.4×10^3	11×10^3
Minimum rigidity C_{min}, kgf per mm		3.25×10^3	2.4×10^3	2×10^3

similar to the ellipse of inertia of a beam cross section in *Strength of Materials.* Upon unsymmetrical bending of a beam, i.e., when the force does not coincide with the direction of the principal axes of rigidity (which are the axes of the ellipse), the direction of the full deformation is perpendicular to a tangent to the ellipse at the point where the ellipse intersects the line of action of the force. There is a corresponding construction for determining the full displacement and its component along the Y axis.

At certain values of the parameters of the system, there may be no displacement of the tool nose along the Y axis or it may be displaced in a direction opposing the projection of the acting force P_y. This corresponds to the zero and negative static characteristic (infinite and negative processing rigidity). The conditions under which such phenomena occur can be found by examining the ellipse of rigidity.

The elastic system of the spindle unit in a lathe includes the flexural system of the workpiece, spindle, spindle bearings, fixture for clamping the workpiece (chuck, centres, etc.) and the torsional system (more accurately, the flexural-torsional system) of the transmission from the motor to the workpiece.

Figure 192 illustrates a diagram of the elastic system when turning a workpiece held in a chuck. In this case, the static link between the flexural and torsional systems is set up due to features of the spindle drive: the torque is developed by force P_n which does not intersect the axis of rotation. Upon twist in the system, this force causes bending of the spindle, and upon bending of the spindle (in the direction of the force) twist occurs. Such a link is typical of gear and belt transmissions.

A similarly linked flexural-torsional elastic system is set up in machining a workpiece between centres with transmission of rotation through a single-end driving dog. In contrast to the preceding system, this system varies its orientation in respect to the tool in the process of rotation.

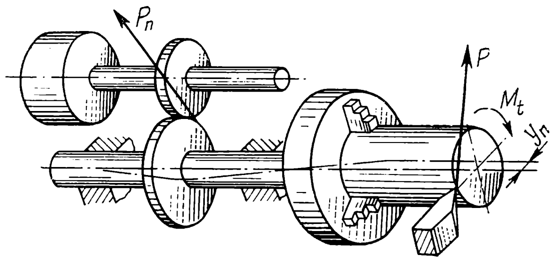


Fig. 192. Diagram of the elastic system of a lathe spindle unit

The described elastic system of the spindle unit may be more complicated if the spindle bearings do not comply with the manufacturing specifications. For instance, if the hole for the front spindle bearing is bored to an oval shape, the rigidity of the bearing will differ in different directions. This must be taken into consideration in an analysis of the elastic system.

For an analysis of the static characteristic it is necessary to measure or calculate the deformations of various units, parts and joints in various directions.

Two methods are employed to assess the obtained data from the point of view of which deformation and the degree to which it affects the relative displacement of the tool and workpiece due to the action of the cutting force. One method is based on the shape of the elastic system after it is changed by deformation while the other is based on the so-called "balance" (structure) of the processing rigidity of the system.

In the first case, the measured or calculated displacements of certain points on the parts or units are plotted in a definite scale on a drawing of the machine tool.

Evaluated in the second case are the relations between the relative amounts of workpiece and tool displacement due to the various types of deformation of the various parts and units of the machine tool. These relations are usually expressed as percentages of the full displacement.

12-4. Working Processes

The inherent stability and the dynamic characteristics of the working processes proceeding in the movable associations of the machine tool are analysed on the basis of data from the corresponding branches of science. Of all the working processes we shall concern ourself with the most impor-

tant—cutting and friction processes—and to a less extent—processes in motors.

The *cutting process* is a complex interrelated system of plastic deformation, thermal processes, friction processes, etc. The cutting process is affected mainly by the cutting tool geometry, cutting conditions (speed, feed and depth of cut), properties of the workpiece material and the cutting (cooling and lubricating) fluid. The principal feature of the element "cutting" is the dependence of the cutting force on the cross section of the undeformed chip or on the cutting speed.

If the cutting force is invariable at a constant undeformed chip cross section and cutting speed, then cutting will be inherently stable.

A breach of inherent stability of the cutting process is manifested as a variation in the force when there is no variation in the size of the undeformed chip cross section or in the cutting speed. Such variation in the cutting force occurs under conditions in which a sheared, discontinuous or segmental chip is formed or in case of an instable built-up edge. The last type of instability is the most important in practice.

The formation of a built-up edge is a significant feature of metal cutting. The built-up edge is formed on the face of the tool beginning with very low speeds. Under certain conditions (of the cutting speed, undeformed chip thickness, etc.), the built-up edge, which is frequently called the stagnant zone, is of a stable nature. In this case, cutting proceeds as if the tool had a rake angle equal to the angle formed by the built-up edge. Chip contraction and the cutting force are sharply reduced and the cutting process is comparatively smooth and uniform. The machined surface does not have a very good finish, but tool life is increased since the tool face is armoured by the built-up edge.

At certain speeds, of a value depending upon the properties of the steel being machined and the geometry of the undeformed chip and of the cutting tool, the built-up edge disappears. In this case, the chip is continuous and a good surface finish is obtained. Here the cutting process is inherently stable.

In a certain speed range the built-up edge is periodically broken off, the cutting force becomes a variable force and the cutting process is inherently instable.

The *static characteristic of the cutting process* expresses the ratio of the cutting force to the undeformed chip thickness obtained in cutting under conditions which are constant with time. At the present time this ratio can be determined only experimentally.

The dependence of the cutting force on the undeformed chip thickness for a given chip width is nonlinear and can be expressed by a power function. Linearization of this function upon variation of the undeformed chip thickness in a small range gives the static characteristic of the cutting process

in the following form

$$K_P = \frac{P}{y} \text{ kgf per mm} \quad (225)$$

where $K_P = Kb$

K = specific cutting force, kgf per sq mm (for carbon steel, $K \cong 200$ kgf per sq mm)

b = width of the undeformed chip, mm.

The *dynamic characteristic of the cutting process* expresses the relation between the cutting force and the undeformed chip thickness for some given variation in the thickness with time.

Practically, the dynamic characteristic of the cutting process can be determined at the present time only by experiment. Dynamic features of cutting were first taken into account in connection with the investigation of vibrations in the cutting process.

A certain time shift exists between a variation in the undeformed chip thickness and the corresponding change in the cutting force. It is due to the inertial nature of the processes in cutting. The inertial nature of thermal processes, associated with the changes in the mechanical properties of metals upon heating or cooling, and of other processes is well known. In the first place it is necessary to indicate the lag in variation of the cutting force which is connected with the limited velocities at which the cut-off volume of material moves from the moment its deformation begins and up to the moment the chip leaves the cutting tool.

In the case of positive tool rake angles, an indice of the degree of deformation of the material in chip formation is the contraction of the chip. On the basis of the general equation for the cutting force, expressed through the principal parameters of the process (undeformed chip thickness, contraction, stresses, etc.) it can be shown that the cutting force varies directly proportional, not to the undeformed chip thickness, but to the actual chip thickness. Upon a change in the undeformed chip thickness, the chip thickness does not change instantaneously, but only after a certain time lag because chip contraction changes in the course of travel of the chip being formed along the face of the cutting tool. This proposition follows from the results of special experiments in which cutting was performed by single-point tools with shortened faces.

The preceding proposition, when taken into consideration, leads to the following simplest form of the dynamic characteristic in cutting (transfer function):

$$W_P = \frac{P}{y} = \frac{K_P}{T_P p + 1} \quad (226)$$

where P and y = variation in the cutting force and undeformed chip thickness, respectively

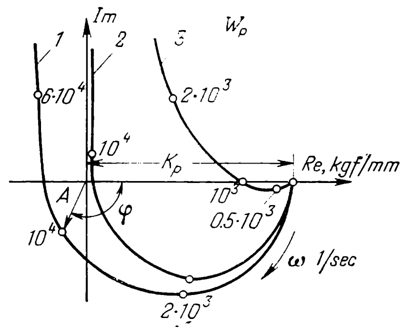


Fig. 193. Gain-phase frequency characteristics of the cutting process for various heights h of the wear area on the tool flank:
$a_0 = 0.1$ mm; $v = 10^3$ mm per sec; 1—$h = 0.1$ mm; 2—$h = 0.5$ mm; 3—$h = 1.0$ mm

$T_P = \frac{m}{n}\,\frac{a_0\xi_0}{v}$ = chip-formation time constant, sec
a_0 = given undeformed chip thickness
$\xi_0 = \frac{a_{10}}{a_0}$ = given chip contraction, equal to the ratio of actual chip thickness to undeformed chip thickness
v = cutting speed
$\frac{m}{n}$ = certain constant factor determined experimentally and equalling approximately 1 to 1.5.

A more complete form of the dynamic characteristic and details concerning its derivation can be found in an article by V. Kudinov called *The Dynamic Characteristic of Cutting*, published in Stanki i Instrumenty (Machine Tools and Cutting Tools), No. 10, 1963. In this characteristic, a significant role is played by the mode of force variation on the tool flank. The three gain-phase frequency characteristics of the cutting process, shown in Fig. 193, were plotted for various height values of the wear area on the tool flank in the orthogonal cutting of carbon steel. At $\omega = 0$, the intercept K_P represents the static characteristic of the cutting process.

It follows from the approximate equation (226) of the dynamic characteristic of cutting that upon a sudden change in the undeformed chip thickness, the cutting force will vary according to an exponential function. The corresponding dynamic time characteristic can be written as

$$P = K_P a\left(1 - e^{-\frac{t}{T_P}}\right) \tag{227}$$

where a = given intermittent variation in the undeformed chip thickness
t = time.

In the dynamic characteristic of the cutting process, an important part is played by the chip-formation time constant. This time constant is reduced with an increase in cutting speed. It increases with the contraction and given undeformed chip thickness.

The time lag of the cutting force leads to a situation in which the cutting force is always less upon a rapid increase in the undeformed chip thickness than upon a reduction.

The *friction process*, like the cutting process, is a complex system of interaction between the most diverse physical and chemical phenomena. Both sliding friction and rolling friction are found in metal-cutting machine tools. The following types of friction are distinguished: dry, boundary, mixed (semidry and semifluid) and fluid.

The processes of dry and boundary friction are determined by extremely complicated and insufficiently examined (as yet) phenomena occurring on the contact surfaces of the bodies. These phenomena are associated with mechanical and molecular interaction of the irregularities on the rubbing surfaces. In mixed friction, the force required to overcome the interaction of the contact surfaces of mating parts is added to the force of viscous resistance of the lubricant which does not wholly separate the surfaces. If the lubricant completely separates the surfaces, we have fluid friction.

In the following, we shall call the different types of sliding friction, associated with contact interaction of surfaces, contact friction. The interaction of surfaces covers discrete contact regions and, upon motion of the bodies, disappears in certain regions and appears in others. The resistance is statistically summed up to obtain the total friction force. This resistance is due to the redeformation of the contacting irregularities.

The inherent instability of the contact friction process is manifested in jamming or seizing phenomena accompanied by destruction of the contacting surfaces to some depth, temperature rise in the friction zone and high instability of the friction force. These phenomena are eliminated by improving lubrication conditions, increasing the hardness of the rubbing surfaces, proper selection of their materials, reduction in the normal load, etc.

The inherent instability of fluid friction is manifested in the transition between laminar and turbulent flow in the lubricant and is rarely found in machine tools.

Since contact friction is most frequently found in machine tools we shall consider this type of friction.

The main laws of contact friction in connection with the deformation of the elastic system are the dependence of the friction force on the normal load (Amontons-Coulomb Law) and on the sliding velocity. Instead of this dependence on the load, a more convenient relationship in making an analysis is the dependence of the friction characteristic on the normal contact deformation. This was introduced into friction investigations by I. Kragelsky.

In this case, the static characteristic of contact friction, in the linearized form, can be written as

$$F = k_f y \tag{228}$$

where F and y = variations in the friction force and normal contact deformation, respectively
$k_f = f\,C_c$ = factor of proportionality between the friction force and the normal contact deformation
f = coefficient of friction
C_c = contact rigidity of the movable association—"friction rigidity".

The static characteristic of friction in respect to velocity, expressing the dependence of the contact or fluid friction on the sliding velocity, can be written in the linearized form as

$$F = k_v v \tag{229}$$

where F and v = variations in the friction force and the sliding velocity, respectively
k_v = factor of proportionality which is positive for an increase in the friction force with an increase in velocity and is negative in the opposite case.

In the case of mixed friction, the characteristic in respect to velocity is of the "drooping" type, as a rule, i.e., the friction force decreases with an increase in velocity. With fluid friction, the characteristic is of the "rising" type.

The dynamic characteristic of the friction process expresses the relationship between the friction force and the normal contact deformation for a certain given variation of this deformation with time. Practically, the dynamic characteristic can only be determined experimentally.

A time shift, due to the inertial nature of processes subject to friction, exists between corresponding variations in the friction force and in the normal contact deformation. It is known that the friction force is developed in the process of the so-called preliminary displacement. The magnitude of the preliminary displacement increases with an increase in the load.

In its simplest form, the dynamic characteristic (transfer function) of contact friction is

$$W_f = \frac{F}{y} = \frac{k_f}{T_f p + 1} \tag{230}$$

where $T_f = \frac{l_f}{v_0}$ = time constant of preliminary displacement
l_f = a certain part of the full preliminary displacement
v_0 = given sliding velocity.

Upon a sudden change in the normal contact deformation (load), the friction force, in accordance with equation (230), will vary along an exponential function. The latter is an approximation of the experimental dependence of the friction force on the preliminary displacement.

The reduction in the friction force with an increase in velocity for mixed friction can be explained by the reduction in contact friction as the body is floated to ride the layer of lubricant by the action of hydrodynamic forces.

The high viscous resistance of the lubricant to the flotation of the body, upon a rapid change in velocity, can be expressed by the simplest dynamic characteristic of mixed friction:

$$W_f = \frac{F}{v} = \frac{k_v}{T_t p + 1} \tag{231}$$

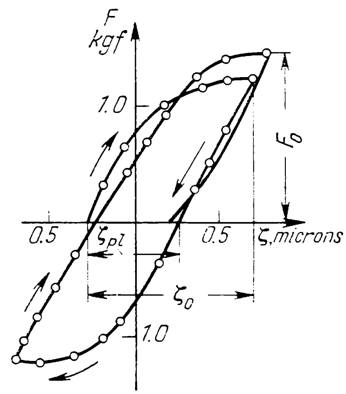


Fig. 194. Experimental dependence of the friction force on the tangential displacement (preliminary displacement ζ)

where T_t is the flotation time constant.

Upon sudden changes in the sliding speed, the friction force varies with a certain time lag according to an exponential function. The lag in time of the friction force from the variations in sliding velocity is the reason for the sharp reduction in the slope of the friction force drop upon a rapid velocity change in mixed friction.

Upon analysing dynamic processes associated with the stopping of travelling parts, or the tool and workpiece (upon reversals, self-excited relaxation vibrations, i.e., self-excited vibrations with stops, etc.), the characteristics of the friction and cutting processes in respect to tangential displacement are of importance. These characteristics are substantially nonlinear. Fig. 194 shows the dependence of the friction force on the tangential displacement. An increase in the displacing force, applied to a body at rest, leads to an increase in the friction force and in the preliminary displacement. A reduction in the displacing force leads to a reduction in the friction force along the other branch of the characteristic since the plastic part of the deformation is not restored in preliminary displacement. Upon changing the direction of the displacing force, which corresponds to a change in sign in the sliding velocity, the effect is repeated in the reverse order.

The characteristic of the cutting process in respect to tangential displacement is of like form. The difference lies in the greater asymmetry of the characteristic which is due to the different geometries of the tool face and flank.

Processes in motors are investigated in such special branches of science as electrical engineering, hydro- and aerodynamics, etc. An idea of these elements of the machine tool dynamic system can be obtained from the simplest dynamic characteristic of the electromagnetic processes in an induction motor:

$$W=\frac{M}{S}=\frac{\frac{1}{\nu}}{T_e p+1} \tag{232}$$

where M = torque of the electric motor
$S=\frac{\omega_0-\omega}{\omega_0}$ = slip
ω_0 = synchronous speed
$\nu=\frac{S}{2M_c}$ = slope factor of the static characteristic
M_c = critical (maximum) torque of the electric motor
$T_e=\frac{1}{\omega_m S_c}$ = electromagnetic time constant of the electric motor
ω_m = angular frequency of the supply mains
$S_c=S_n\,(K_m+\sqrt{K_m^2-1})$ = critical slip
S_n = nominal slip
K_m = maximum-to-rated torque ratio.

The static characteristic of an induction motor, in the form of the so-called mechanical characteristic, is well known.

12-5. Analysis of Dynamic Performance of Machine Tool Systems and the Calculation of Performance Indices

Frequency methods, employed to analyze the dynamic system of a machine tool, are very convenient, both in their comparative simplicity and in the wide opportunities presented for using experimental data.

The dynamic calculations of a machine tool system consist in plotting the gain-phase frequency characteristics (GPFC) of the equivalent elements (EES and MS) and the complementary system of elements (working processes), linked to the equivalent elements, followed by an analysis of a one-loop system of one of the types indicated in Sec. 12-1. The selection of the type of system depends upon nature of the problem to be solved.

If the characteristics of the elements have been obtained experimentally, for example by testing pilot models, the indices of dynamic performance are analyzed in the same way as in designing a new machine tool.

In many cases, it proves more convenient, when determining the GPFC either by calculations or experimentally, not to consider its elements separately, but to find the characteristic of the corresponding open-circuit system, characteristic of a closed-circuit system with a given external action (input), etc. Due to the extremely complex nature of machine tool dynamic systems, such calculations are practically feasible only if an electronic computer is available.

Before determining the characteristics of the elastic system and the working processes, it is necessary to assess the inherent stability of each one.

Calculations of a system in which one of the elements of the machine tool dynamic system is inherently instable involve the solution of the nonlinear problem for determining the amplitudes and frequencies of self-excited vibrations. This is followed by an estimation of the permissible level of these vibrations. An example of such calculations is the determination of the self-excited vibrations in an inherently instable cutting process, i.e., in the formation of discontinuous or segmental chips, or chips when a built-up edge breaks off periodically. The procedure for solving such problems is similar to the one considered below for calculating self-excited relaxation vibrations.

The linear approximation of the system is analyzed on the basis of the characteristic of a disconnected one-circuit system of one of the three indicated types. The characteristic of a disconnected system is plotted as the product of the characteristics of the component elements which make up the system. For each value of the frequency, the amplitudes are multiplied together while the phases are added together. This will provide a dimensionless characteristic. Shown in Fig. 195, as an example, is a disconnected system of the third type and its characteristic W_{dis}, plotted for the case of a machining operation "without previous chatter marks". Thus

$$W_{dis}=\frac{y'}{y''}=W_{EES}W_P \tag{233}$$

In analyzing the system for the case of machining "along chatter marks", the disconnection is made along the delayed feedback as shown in Fig. 196. The characteristic W_y of the system from which the element with time lag has not been separated is the characteristic of a closed-circuit dynamic system for machining "without previous chatter marks" taken in reference to the external action on the working process. Thus

$$W_y=\frac{y}{y\,(t)}=\frac{W_{dis}}{1+W_{dis}} \tag{234}$$

This characteristic is plotted either after it is calculated according to the known characteristic W_{dis} or graphically by using the relationships for the

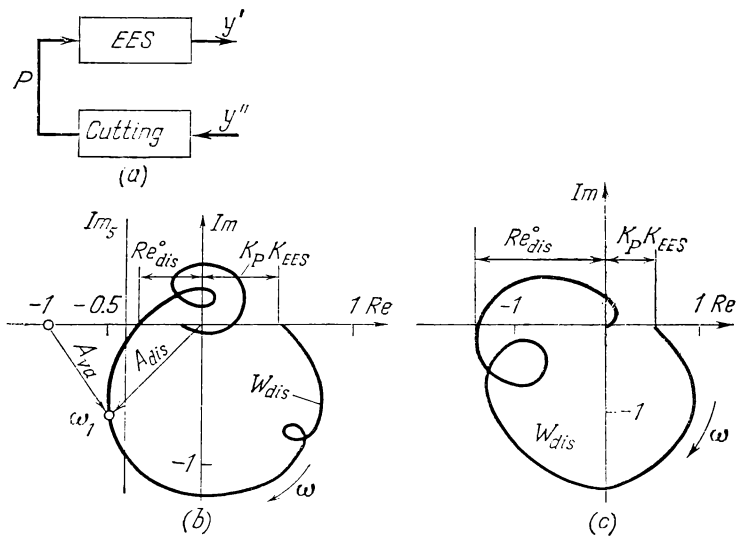


Fig. 195. Disconnected equivalent dynamic system (in respect to cutting) and its gain-phase frequency characteristics:
(*a*) block-diagram; (*b*) stable; (*c*) instable

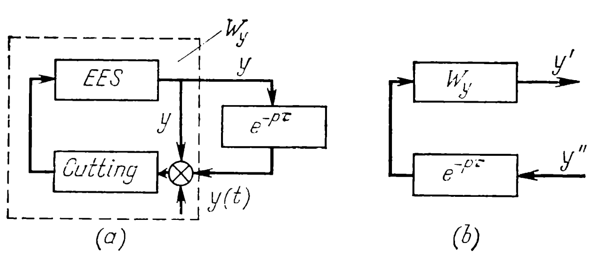


Fig. 196. Block-diagrams of an equivalent dynamic system (in respect to cutting) in machining "along chatter marks":
(*a*) closed circuit; (*b*) disconnected

amplitude and phase:

$$\left.\begin{array}{l} A_y = \dfrac{A_{dis}}{A_{va}} \\ \varphi_y = \varphi_{dis} - \varphi_{va} \end{array}\right\} \qquad (235)$$

where A_y and φ_y = amplitude and phase of characteristic W_y of a closed-circuit system in respect to action $y(t)$

A_{dis} and φ_{dis} = amplitude and phase of characteristic W_{dis} of a disconnected system

A_{va} and φ_{va} = amplitude value (module) and phase (independent variable) of the vector drawn from point (— 1, $i0$) to the point of characteristic W_{dis} with a given frequency value (this vector is shown in Fig. 195).

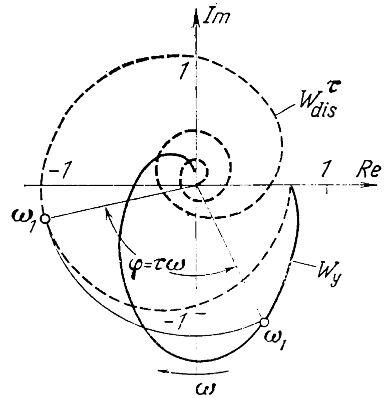


Fig. 197. Constructing the characteristic W^{τ}_{dis} for a given time lag

The characteristic of a disconnected system in operation "along chatter marks" can be written as

$$W^{\tau}_{dis} = W_y e^{-\tau p} \qquad (236)$$

where e = base of the natural logarithms

τ = given time lag.

The plotting of this characteristic for a system with delayed feedback is illustrated in Fig. 197. It consists of turning the vectors of the characteristic through the angle $\varphi = \tau\omega$. The dynamic performance indices of a machine tool system can be analyzed using a simplified system of a machine tool in the process of machining. The procedure and principal results are completely valid for two other simplified systems used in analyzing the drive or machine tool idling operation. In respect to the last system, we shall concern ourselves only with certain specific problems associated with features of the travel of machine tool units along ways.

Margin or Degree of Stability of a System

In general case, a system is considered stable if its deviation from a given state (equilibrium, or motion according to a given law) in the transient process, caused by limiting the action in value, decreases with time. If the deviation increases, the system is considered to be instable.

The nonlinearity of the system, i.e., the variation of its parameters with the deviation, gives rise to a situation in which the deviation does not increase without limit, stopping when it reaches a certain value.

Upon periodic instability, oscillations of a certain amplitude are initiated. They are called self-excited oscillations.

In practice, self-excited oscillations in metal cutting are usually called vibrations or chatter, and the stability of the dynamic system of the machine tool in cutting is called its vibration-proof properties.

The complexity of a machine tool dynamic system, in which the parameters vary in very wide ranges (the cutting speed, for instance, may vary by two or three orders of magnitude) renders impossible all attempts to explain the origination of self-excited vibrations in the cutting process by the action of any single "exciter".

A manifestation of the aperiodic instability of a machine tool dynamic system is the gouging or dig of single-point tools. The deviation (divergence) of the tool or workpiece, occurring in this type of instability, increases continuously with time due to the deformation in a single direction. The tool cuts deeper and deeper into the metal, the cutting force increases, leading to a further increase in deformation. Thus, gouging or dig, originated by an accidental impact, ends in breakage of the tool or workpiece.

The *stability of a machine tool dynamic system* is estimated on the basis of the extent of the so-called *stability range* in space of the parameters of the system or, in other words, of the limits of variation of the parameters within which the system does not lose its stable state. For example, the stability range in boring a hole with a stub boring bar is limited by the overhang of the bar, usually equal to 4 or 5 bar diameters (this value may vary in accordance with other parameters: cutting speed, workpiece material, etc.). It is necessary that the working range of variation of the machine tool parameters be within the limits of the stability range of the system.

The stability of a system can be determined either by calculations or experimentally.

In these calculations, the differential equation of the machine tool dynamic system is analyzed. If the solution of the equation increases with time, the system is instable. In the majority of cases in practice, however, the equation is not solved, and the stability is estimated by means of the so-called stability criteria which enable a result to be obtained with comparative ease for a linearized equation. Criteria may be either of the algebraic (Routh, Hurwitz, etc.) or frequency type (Nyquist, Michailov, etc.).

More detailed information on stability criteria can be found in any textbook on automatic control theory.

The Nyquist gain-phase criterion can be applied to estimate the stability of a system. It enables the effect of the elastic system and the cutting process to be readily followed out on the characteristic of a disconnected system.

In its simplest interpretation, this stability criterion can be reduced to the following: if the characteristic cuts off an intercept $Re^{\circ}_{dis} < |1|$ on the negative real axis, the system is stable for the given values of the parameters. If the characteristic of the disconnected system intersects the negative real axis beyond —1, the system is instable. When the intercept Re°_{dis} is equal to unity, the system is on the boundary of stability and the intercept can be used to estimate the limiting width b_{lim} of the undeformed chip permitted by the system on the basis of stability conditions.

The larger the intercept Re°_{dis} cut off on the negative branch of the real axis, the less the limiting width of the undeformed chip and the lower the vibration-proof property of the system. This rule is in good agreement with another: *the smaller the intercept $K_P K_{EES}$ cut off by the characteristic at $\omega = 0$ on the real axis, the higher the processing rigidity of the system and the less the effect of deformation of the system on machining accuracy.* If $K_P K_{EES}$ has a negative value exceeding $|1|$, gouging or dig occurs.

The frequency of self-excited vibrations, originated upon a loss of stability, is close in value to the frequency of characteristic ω_n at the point where it intersects the negative real axis.

The physical concept of the loss of stability in cutting "without previous chatter marks" consists in the following. Because of the many degrees of freedom of the elastic system, the vibration of the tool in respect to the workpiece is the result of adding several interlinked simple oscillations. Hence, the path of relative motion of the tool and workpiece, due to this addition of oscillations, has the shape of a closed curve, approximating an ellipse. In contrast to the rigidity ellipse, this is called the displacement ellipse. Motion of the cutting tool along such a path varies the undeformed chip thickness and, consequently, the cutting force in such a manner that upon tool motion in the direction of cutting force action, the undeformed chip thickness will be larger than upon tool motion opposite to the cutting force.

The conditions under which such motion of the system occurs conform to the case shown in Fig. 189 in which the normal forms of vibrations of different signs are added together. In this case, the characteristic W_{EES} intersects the negative real axis and the system is potentially instable.

If the system is stable, the phase shift between the vibrations is such that the direction of motion of the tool nose is the reverse of that described above. The variation of the cutting force under these conditions has a damping effect on the vibrations, not replenishing the energy being dissipated, as in an instable system, but increasing its dissipation. Special note should be taken of this circumstance, since literature on the subject gives no clear idea of the effect of cutting (in a stable system and in the absence of vibrations) on the vibrations caused by external disturbances.

It should be noted that in the case being considered, the variation in the damping effect of cutting is determined, not so much by cutting itself, as by

the variations in the elastic system which determine the direction of motion along the path or, what comes to the same thing, the stability of the system.

It was mentioned above that the behaviour of a stable closed-circuit system upon external actions differs from the behaviour of an open-circuit system. The damping effect of cutting is one of the factors illustrating this proposition.

The role, described above, played by the variation in undeformed chip thickness in initiating self-excited vibrations is supplemented by the effect of the lag in the cutting force in respect to variations in undeformed chip thickness. In other words, in self-excited vibrations, ambiguity of variations in the cutting force is the result of the time lag of the cutting force in respect to the variations in undeformed chip thickness.

The dependence of the dynamic characteristic in cutting (the chip-formation time constant) on the velocity determines whether there are two or more boundary cutting speeds, above and below which the system is stable and no vibrations occur. The occurrence of such cutting speeds which limit the region in which vibrations are initiated has long been known to investigators and in practice.

It must be noted, however, that in the given case, this concerns the values of the parameters of the system at the stability boundaries, and not the amplitude values of self-excited vibrations which depend upon the character of the nonlinearity limiting their increase.

In machining "along chatter marks", the stability of the system is sharply reduced. This follows from an analysis of the characteristic of a disconnected system with delayed feedback. Regardless of the time lag, the system will be stable if $A^{\tau}_{dis} < 1$. Since $A^{\tau}_{dis} = A_y$, this condition, in accordance with equation (235), is complied with when the characteristic W_{dis} of the disconnected system is to the right of the straight line Im_5, drawn parallel to the imaginary axis through point (—0.5, $i0$) as shown in Fig. 195. In the case of high-speed machine tools operating with multiple-edge tools and in grinding, the stability range can be extended for certain cutting conditions (speeds, feeds and depths of cut). The possibility of this extension is determined by turning characteristic W_y, in plotting W_{dis}, in such a manner that it does not encompass point (—1, $i0$). In such cases, the time lag is small and near to the period of vibrations of the system.

If, however, the time lag is large, as in turning steel parts of medium and large sizes, this possibility (of extension) is practically excluded.

It will be recalled that the time lag in the last case is equal to the time for one revolution of the workpiece.

In machining with a multiple-edge tool

$$\tau = \frac{60}{nz} \tag{237}$$

where z = number of cutting edges (cutter teeth, etc.)
n = tool speed, rpm.

The special feature of grinding, mentioned in Sec. 12-1, is the effect of wheel wear on the dynamic properties of the system. Primarily, this is manifested in the variation of the time lag. A short time after beginning work (and practically during truing, insofar as the wheel is concerned) waves appear on the surfaces of the workpiece and wheel due to forced vibrations. From the point of view of the workpiece, the wheel becomes a kind of "milling cutter" with a number of "teeth" equal to the number of waves, while the workpiece becomes a sort of "broach" which wears down the wheel.

The lag time in this case is determined by the following equations:

(1) in feedback in respect to the "chatter marks" on the workpiece

$$\tau_{wk} = \frac{60}{n_{wh} z_{wh}} \tag{238}$$

where n_{wh} = wheel speed, rpm
z_{wh} = number of waves on the wheel periphery;

(2) in feedback in respect to the "chatter marks" of wheel wear

(a) for cylindrical grinding

$$\tau_{wh} = \frac{D_{wk}}{D_{wh}} \frac{60}{n_{wh} z_{wk}} \tag{239}$$

(b) for surface grinding

$$\tau_{wh} = \frac{60 t_w}{\pi D_{wh} n_{wh}} = \frac{v_t}{f \pi D_{wh} n_{wh}} \tag{240}$$

where D_{wk} and D_{wh} = diameters of the workpiece and wheel, respectively
z_{wk} = number of waves on the workpiece periphery
t_w = wave pitch on the surface being ground
v_t = table (or work) speed in surface grinding
f = frequency of vibrations (frequency of waves on the workpiece surface).

Thus, machining "along chatter marks" reduces the stability of the system by one half, at best, and by considerably more in ordinary cases.

The most convenient parameter for estimating the stability in calculations or experiments is the limiting width of undeformed chip that can be removed on the machine tool without danger of vibrations and dig. The boundary of stability can be expediently plotted in the co-ordinates "limiting width of undeformed chip vs cutting speed (rpm)" at various feeds since these parameters determine the output in machining. By comparing the stability boundaries we can make a relative estimation of the performance of different specimens of the same machine tool model, different models, different constructions of cutting tools or fixtures, or different sets of cutting conditions.

It should be noted, however, that the stability boundaries characterize the margin or degree of stability of the system only indirectly.

In calculations, the margin of stability can be conveniently estimated on the basis of the length of the intercept (in machining "without previous chatter marks")

$$(1 - Re^{\circ}_{dis})\,100\% \tag{241}$$

and of the phase angle through which characteristic W_{dis} is turned until it intersects the negative branch of the real axis at the point with the co-ordinates $(-1,\ i0)$.

A calculated, and mainly an experimental, estimation of the machine tool system can be conveniently made, employing the stability factors of a closed-circuit system in respect to external actions. These factors are taken for a potentially instable form of vibrations of the system (i.e., the form that occurs when the system loses its stability). These vibrations are characterized by the value of the natural frequency ω_n.

The stability factors are of the forms: (a) in respect to external action on the EES (A_{va}) and (b) in respect to external action on the cutting process $\frac{A_{va}}{A_{dis}}$.

The procedure for determining A_{va} and A_{dis} has been described above. For making an experimental estimation of these factors, data are used concerning the vibrations in cutting and the vibrations during an idle run or periodical variations in the machining allowance. The corresponding equations will be given below in considering forced vibrations.

It may prove convenient in many cases to make an estimation of the machine tool system on the basis of how rapidly the stability factor varies with the variation of some parameter of the system, for instance, the depth of cut, feed, boring bar overhang, etc.

Let us consider methods of improving the vibration-proof properties of a system.

The role of the elastic system is completely determined by its dynamic characteristic which depends upon the rigidity, mass, damping ability and interaction of the various oscillating circuits that make up the system.

The direction of the action on the elastic system can be determined by applying a gain-phase stability criterion. To raise the stability, it is necessary to reduce the radius vector of its dynamic characteristic, especially in the regions adjacent to the negative real axis, and also to ensure such an arrangement of the characteristic in which it does not intersect the negative real axis.

One of the principal methods employed in practice to improve the vibration-proof properties of a system is to increase its rigidity, as a result of which the radius vector of the characteristic is correspondingly reduced.

As a rule, errors in manufacturing the parts and in assembling all the elements making up the machine tool elastic system lead to a reduction in the stability of the system. One of the most common and essential errors in the manufacture of machine tools with a rotary working motion (lathes, milling machines, etc.) is out-of-roundness (ovality) of the bore in which the spindle bearing is installed. The rigidity of the spindle bearings, in this case, differs in different directions. This creates a co-ordinate interlink in the system, sharply reducing the stability.

In most cases, out-of-roundness in the bore is a result of errors of the boring machine in which the machining was done. In such cases the equipment is realigned or relevelled and the bore is machined a second time. As a last result, the outer ring of the bearing can be locally metalized to obtain an oval form and then fitted to the bore.

Sometimes distortion in the shape of the bore is due to deformation of the headstock housing when incorrectly located fastening bolts are tightened.

Nonlinearities in the form of clearances and backlash are found especially frequently in inaccurate manufacture and adjustment of spindle bearings and in kinematic trains. Excessive clearances in spindle bearings substantially deteriorate the vibration-proof properties of the system.

If the system is sufficiently rigid, vibration can be eliminated both by lowering and raising the cutting speed. If the speed is increased, the possibility of the occurrence of vibrations of higher frequency should be considered. For example, in boring with a bar in a lathe, vibrations of three frequencies—about 500, over 1000 and over 10,000 cps—appeared consecutively as the cutting speed was varied. The first frequency turned out to be near to the flexural oscillations of the boring bar, the second to the torsional oscillations and the third to the flexural oscillations of the boring bit.

In many cases, vibrations can be efficiently eliminated by simply changing the cutting speed.

The effect of the rate of feed depends upon the speed range employed in the machining operation; the vibration-proof properties may be either improved or deteriorated with an increase in feed. In roughing or semifinish machining on machines of the lathe type, when a high-speed steel or carbide-tipped tool is used, an increase in the rate of feed will usually contribute to the elimination of low-frequency vibrations.

An increase in the depth of cut in a lathe, which determines the width of undeformed chips of rectangular cross section*, always leads to the initiation and intensification of vibrations. Reducing the depth of cut to eliminate low-frequency vibrations is a simple measure but leads to a substantial reduction in labour productivity and can thus be resorted to only in certain cases.

* In turning with a tool having a 90° approach angle.

It is sometimes expedient to change the cutting conditions so as to transfer to a different form of vibrations whose elimination can be accomplished more simply by another method, for instance by action affecting the elastic system.

Measures for eliminating tool gouging and dig are simple in principle and consist in properly setting up the single-point tool, setting the tool upside down, reducing tool overhang, and increasing the rigidity of the elastic system and reducing, at the same time, the difference in the principal rigidities.

An analysis of the machine tool dynamic system in idle-run operation, carried out in connection with the selection of parameters for the drive, elastic system or friction, does not in principle differ from that described above. A popular opinion held by specialists and found in literature on the subject is that the "drooping" characteristic of friction in respect to velocity or the difference between the forces of static friction and of friction of motion has a considerable influence on the stability of travel of parts in a machine tool. The second of these factors shall be treated below.

Investigations show that the "drooping" characteristic of the friction forces, in the same way as the cutting forces, though they have a considerable effect on the mean level, especially of the friction forces, have no significant effect on the stability of the system. This can be explained by the inertial nature of the processes determining the characteristic, for instance, the process of flotation in mixed friction. This inertial nature reduces the slope of the characteristic to a great degree and correspondingly reduces the variation in the friction and cutting forces upon variations in sliding or cutting speeds.

In analyzing the stability of the system "at large", required in connection with the possibility of relaxation in idle travel of the units ("stick-slip" phenomena of sliding units at low speeds) or in cutting, of main significance is the nonlinear dependence of the forces on the tangential displacement described in Sec. 12-4. The difference between the forces of static friction and friction of motion, which is small in comparison to the magnitude of the friction force and may be absent in vibrations, is not the governing factor in the initiation of self-excited relaxation vibrations.

Self-excited relaxation vibrations are identified by the following condition based on the presence of stops:

$$A \geqslant \frac{v}{\omega} \tag{242}$$

where A = amplitude of the vibrations
ω = natural frequency of the system
v = velocity of the given motion (cutting or sliding).

These vibrations may be of two types: (a) without change in the sign of the velocity when the amplitude of the vibration of velocity $A\omega$ is equal

to the given velocity of motion v, and (b) with a change in the sign of the velocity, when $A\omega > v$.

Upon vibrations of the first type, the force varies along the first half of the hysteresis loop shown in Fig. 194. Upon vibrations of the second type, the second half of the loop is covered. This, in contrast to the first half, characterizes the variation in the force which counteracts the development of vibrations. If the two halves of the loop are equal, a feature characteristic of friction forces, there can be only one self-oscillating regime with an amplitude that does not vary with a variation in velocity, and a frequency that increases with an increase in velocity.

The characteristic is symmetrical for vibrations of the first type and has the same shape as the characteristic of a clearance with a limiting feature.

Approximate calculations are carried by the describing function method of nonlinear mechanics. In employing this method, a closed-circuit system is represented as consisting of two elements: linear and nonlinear. In the case in which the characteristic W_{EES} of the equivalent elastic system can be represented by a single normal form, the characteristic W_l of the linear part is of the third order and equal to the product of W_{EES} and characteristic W_ξ of preliminary displacement. The latter determines the formation of preliminary displacement in friction contact or in the cutting zone upon deformation of the equivalent elastic system. Fig. 198 shows the characteristic of the linear part and its two component characteristics. Also shown is the inverse equivalent characteristic of the nonlinear element which expresses the dependence of the describing function on the amplitude.

The intersection of this characteristic, taken with the reverse sign, with the characteristic of the linear part gives the self-oscillating regime. A solution satisfying stability conditions corresponds to point d of the intersection. The amplitude of the self-excited vibrations is determined from the characteristic of the nonlinear element; the frequency from the characteristic of the linear part.

Upon an increase in the velocity of motion, the condition for the occurrence of self-excited vibrations of the first type requires an increase in the amplitude. As the amplitude increases, there is a corresponding increase in the height of part of the first half of the hysteresis loop, i.e., the magnitude of the variation in the force of friction or in cutting upon vibrations. The amplitude reaches its maximum value at the full height of the hysteresis half-loop, i.e., when it is equal to the whole force of friction or cutting. This condition serves as the basis for determining the maximum velocity of travel up to which self-excited relaxation vibrations of the first type can exist at a given value of the force and a given characteristic of the equivalent elastic system. The frequency of relaxation vibrations of the first type remains practically constant upon variations in the velocity of travel.

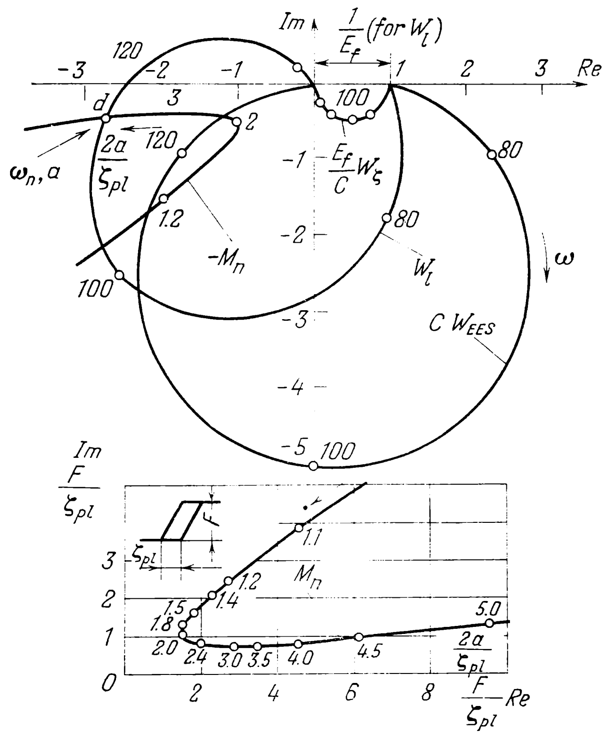


Fig. 198. Gain-phase frequency characteristics W_l of the linear part of a system and its elements; nonlinear element—M_n (inverse equivalent characteristic)

Figure 199 shows the variation in the amplitude and frequency of self-excited relaxation vibrations with the travel velocity. The curves were plotted to data calculated by the above-described method.

The dependence of the time constant of preliminary displacement on the travel velocity makes it possible for a lower boundary velocity of self-excited relaxation vibrations to exist, besides the upper boundary velocity. Self-excited relaxation vibrations are absent at velocities of travel below the lower boundary velocity.

To eliminate self-excited relaxation vibrations or to reduce the range of their occurrence, it is necessary to reduce the value of the friction force or cutting force by proper selection of the materials and lubricant, by relieving the load on the friction surfaces, by reducing the undeformed chip width, etc. Measures affecting other parameters of the nonlinear characteristic.

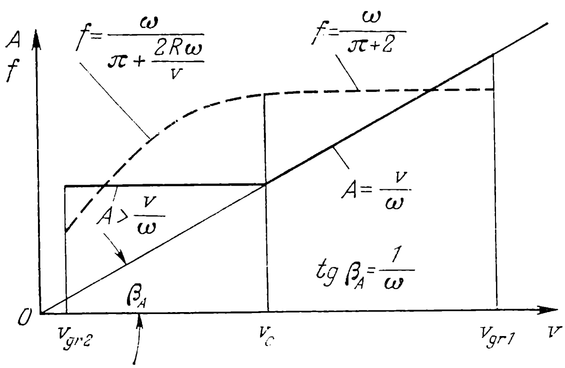


Fig. 199. Variation in the amplitude A and frequency f of self-excited relaxation vibrations with the travel velocity

such as a reduction of its slope, reduction of the permanent set, i.e., width of the loop, etc., are of advantage. An increase in rigidity of the system and the damping of the potentially instable normal form of vibrations of the EES comprise one of the most important practical methods for eliminating self-excited relaxation vibrations.

The application of special lubricants with active additives, such as the Soviet lubricant, grade ВНИИНП-401, can greatly increase the stability of travel of parts and units of machine tools. Various methods of eliminating self-excited relaxation vibrations in cases of small movements or displacements were investigated by V. Push.

Methods of raising the degree of stability "locally" upon travel of the machine tool units are similar to the methods described for improving the vibration-proof properties and consist in eliminating the interlinks in the equivalent elastic system, increasing the rigidity of potentially instable forms of the EES, reducing the coefficient of friction, etc.

Deviations of the system upon external actions. These deviations are the most important indices of the dynamic performance of a machine tool since they determine the possibility of achieving the specified machining accuracy under conditions in which the dynamic system is in a stable state at maximum production capacity.

Let us consider the behaviour of the closed-circuit dynamic system of the machine tool in the process of cutting upon external actions, i.e., actions that do not vary upon deformation of the system.

These actions can be divided into two groups:

1. External force and kinematic actions on the EES, denoted in Fig. 184 by the arrow f (t).

2. External actions on the working process—cutting—and denoted in Fig. 184 by the arrow $y(t)$. In the following, we shall call this group changes in adjustment.

The first group includes:

(a) periodic forces and impacts transmitted to the foundation of the machine tool through the soil from various extraneous sources of disturbance (traffic, operation of power hammers, compressors, etc., in the immediate vicinity of the machine tool);

(b) periodic forces originating from unbalance of rotating components (rotors of electric motors, grinding wheels, blanks, etc.);

(c) periodic forces originated due to errors in toothed gearing, nonuniformity of belts and the presence of belt joints, errors in spline and key joints, misalignment of couplings and clutches, nonuniformity of the rolling members or waviness of the races in ball and roller bearings, etc., as well as the pulsating loads of pumps in the hydraulic and lubrication systems, etc.;

(d) variable cutting forces resulting from inhomogeneity of grinding wheels, workpiece material, etc.;

(e) variable inertia forces developed in the reversal of tables, rams, slides and other units in machine tools with reciprocating motions;

(f) periodic forces of ultrasonic frequency, artificially produced to improve the cutting process and to obtain a higher class of surface finish on the machined surfaces.

A feature of this group of actions, with the exception of the last one, is that they mainly have a detrimental effect on the quality of machining and on the service life of the machine tool. Hence, efforts should be made, first of all, to eliminate them. Methods of reducing their effect on the quality of the workpiece will be mentioned below.

The second group includes:

(a) variability of the undeformed chip cross section in milling;

(b) variability of the undeformed chip cross section in machining interrupted surfaces or blanks with a variable machining allowance (such conditions most frequently occur in heavy machine tools in machining forgings or castings);

(c) variability of the undeformed chip cross section as the tool feeds in and runs out of the workpiece (single-point tools, twist drills, core drills, shaping cutters, etc.);

(d) variability in the magnitude or direction of the feed motions in tracer- and numerical-control machine tools;

(e) variability of the cutting speed in facing in a lathe or in turning noncircular parts (castings of square cross section).

All of these causes are associated with features of the manufacturing process and it is practically impossible to eliminate them (with the exception of the variability of machining allowances which should be reduced

by improving the techniques of blank manufacture). Hence, it is necessary to reduce their effect on the quality of machining.

All the above-listed causes lead to the initiation of complex forced vibrations of the system or aperiodic displacement of the units in the course of machine tool operation. It is not always possible, from the mode of these vibrations or movements, to find the cause of their initiation.

The following methods are used to reveal the source of an influence acting on the machine tool:

1. Frequency analysis of the vibrations of the system and a comparison of their frequencies with the frequencies of possible sources of the disturbances (speed of the rotor of an electric motor, grinding wheel speed, frequency of pulsations of a pump, the number of times the rolling members in a bearing pass through the load zone, the number of meshing movements of mating gears, number of entries of milling cutter teeth into the workpiece, etc., in a unit of time). It should be noted that the nonlinearity of the system and the complex nature of the disturbances lead to the initiation of forced vibrations, not only with the main frequency of the disturbance, but also with the natural frequencies of the system.

2. The switching off, removal or replacement of possible sources of disturbance followed by an analysis of the results of this measure. It is possible, for instance, to switch off the electric motor, pumps or spindle rotation; to replace bearings, gears or a milling cutter (by a cutter with a different number of teeth), etc.

To reveal sources of disturbance which are transmitted through the foundations, tests are conducted when neighbouring shops are not in operation and there is no traffic, etc., for example, at night or a day of rest.

The second method is the one mainly employed in practice, but it requires a large labour input and does not always provide the desirable results at once. It proves much more expedient to combine this method with the first method, even though this requires the application of vibration measurement apparatus. After revealing the source of disturbance, measures are taken to reduce its effect on the machine tool dynamic system. In some cases it is possible to completely eliminate the source of disturbance, for instance, to move a railroad line farther from the shop, to rearrange the equipment in the shop, to replace a bearing having a wavy ball or roller race, etc. If this cannot be done, measures are taken to reduce the intensity of the disturbance. Such measures include: balancing rotating parts (rotors, grinding wheels, etc.), using tools (milling cutters, broaches, etc.) providing a more gradual variation of the undeformed chip cross section, using a tooth with a pitch of which the width of the surface being machined is a multiple, increasing the smoothness of reversal of the reciprocating units, etc. When the possibilities indicated above have been exhausted, measures are resorted to isolate the system from the sources of disturbance.

A large number of different antivibration devices have been developed. They are chiefly used to isolate the system from disturbances transmitted through the soil or foundation plate of the shop, as well as disturbances initiated by electric motors and hydraulic systems.

In the first case the machine tool is installed on a special foundation comprising a massive concrete cube suspended on springs. The mass of the foundation and the rigidity of the springs are selected from the condition that the natural frequency of vibrations of this system should be remote from the natural frequencies of the system, determined chiefly by the relative displacements of the tool and workpiece, while the deflection of the springs under the action of the variable weight load (due to traversing the units or shifting the workpiece on the machine) should not exceed a certain permissible standard value.

To isolate the machine tool from disturbances transmitted by the foundation plate of the shop, it is sometimes sufficient to separate the machine foundation from the plate with a layer of sand, cinders, cork or other materials having a high damping capacity.

An extensively used method is the installation of the machine tool on antivibration pads or mats of rubber, felt or special synthetic materials, as well as on shoes of special construction with shock-absorbing properties.

Electric motors are also installed either on shock-proof plates or antivibration pads. In many cases, best results are obtained by a careful fitting of the jointing surfaces of the machine tool and electric motor.

A realization of all the measures described above leads in the final analysis to the elimination of some disturbances and the reduction of others. However, the variability of adjustments and certain other types of external influences remain, and it is necessary to reduce their effect on the quality of machining.

The diverse conditions of machining and of types of external actions require that the reaction of the machine tool dynamic system, in the form of relative displacement of the tool and workpiece, be estimated in each case.

Next we shall consider *machining errors* which occur in connection with the following principal types of external actions:

1. Actions that are constant with time and produce the static machining errors.

2. Periodic actions that produce forced vibrations and the corresponding stationary dynamic machining errors in the form of waviness, lobed-shape, etc.

3. Influences in the form of rapid changes of magnitudes (undeformed chip cross section, forces, etc.) from one steady-state value to another and producing transient dynamic error.

The *static error* determines the magnitude of the machining error, due to deformations of the system at constant cutting conditions. Very conven-

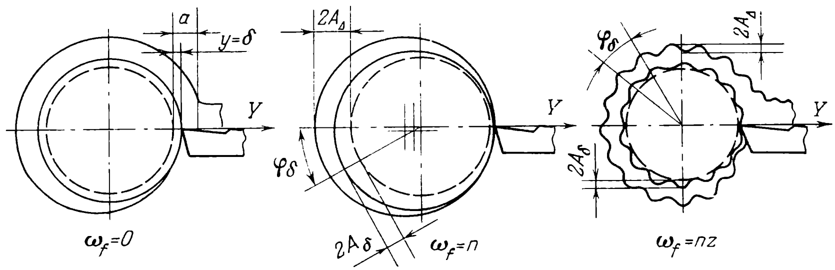


Fig. 200. Machining errors caused by variations in the machining allowance

ient in estimating the static error is "accuracy improvement" which was proposed by K. Votinov and applied in process engineering theory by A. Sokolovsky.

"Accuracy improvement" is the ratio of the like errors of the blank Δ and the machined workpiece δ, shown in Fig. 200. In turning a shaft, the variation in the machining allowance agrees with the variation in diameter, out-of-roundness of the blank agrees with the out-of-roundness, taper of the blank agrees with the taper of the machined workpiece, etc.

From the point of view of the processing engineer, "accuracy improvement" enables a relationship to be established between machining accuracy and cutting conditions. It is possible, in this case, to determine the cutting conditions (speed, feed and depth of cut) which will ensure that the specified accuracy is obtained, or determine the machining accuracy attainable with given cutting conditions. The same accuracy improvement is used to determine the effect of the processing rigidity of the system on the machining accuracy.

In accordance with equation (234), where $y = \delta$ and $y(t) = \Delta$, we obtain for $\omega = 0$

$$\delta = \Delta \frac{K_P K_{EES}}{1 + K_P K_{EES}} \tag{243}$$

and accuracy improvement

$$\frac{\Delta}{\delta} = 1 + \frac{1}{K_P K_{EES}} \tag{244}$$

It is known that in machining workpieces of homogeneous material with a constant machining allowance, the effect of the deformation of the system or, in other words, the static error of the system can be compensated by suitable adjustments. However, if the rigidity of the system varies during the machining operation, for example along the length of a shaft being

turned or with a change in the overhang of a stub boring bar, an additional machining error appears. This error must be eliminated, in the same way as the error due to a variable machining allowance or variable hardness of the material, by machining in several passes or by suitable selection of the rate of feed.

A direct functional relationship between the processing rigidity (static characteristic of the elastic system) and machining accuracy is almost undetectable because of the influence of other factors besides the deformability of the system. This relationship exists, however, in a more complex form, the so-called correlational link.

Errors in the case of forced vibrations appear on the machined surface in the form of errors in shape (lobed-shape, etc.), waviness or microirregularities depending upon the ratio of the size of the surface to the wave pitch, as well as on the direction of the formative motion of the tool and the direction in which measurements are made.

Given the gain-phase frequency characteristic of the disconnected system, from which the stability in cutting is determined, there is no difficulty in estimating the variation in the amplitude of the forced vibrations as a function of their frequency and the stability of the system. It is important to differentiate between the two groups of external actions mentioned above: force or kinematic actions and changes in adjustment (in the given case these are vibrational).

Data on actions of the first group are usually available in the form of the frequencies and amplitudes of vibration of the tool and workpiece, i.e., of the elastic system of the machine tool, under conditions when the source of disturbance is active but no cutting is being done. Such are vibrations measured during idle-run operation of the machine tool, vibrations from the foundation measured when the machine tool is switched off, etc.

Data on actions of the second group are available in the form of the amplitude of the variations in undeformed chip thickness and the frequency of these variations. Of this kind, for example, are data on the variability of the undeformed chip in milling, in turning an eccentric blank, etc.

In cutting, when the machine tool dynamic system becomes of the closed-circuit type, the above-indicated amplitudes of the vibrations are changed.

The amplitude A_f *of forced vibrations in cutting or the amplitude of the waves on the workpiece surface due to external force or kinematic actions* is equal to the amplitude A_{ir} of vibrations of the elastic system when no cutting takes place, divided by the stability factor in respect to external action on the EES. This factor is determined from the gain-phase frequency characteristic of the disconnected system as the amplitude value (module) of vector A_{va}. Thus

$$A_f = A_{ir} \frac{1}{A_{va}} \tag{245}$$

The amplitude A_{ir} of vibrations of the EES due to external actions is determined in measuring the vibration level of the machine tool on an idle run (separating out the given harmonic component) or by calculating the forced vibrations of the EES due to the specified external action. These calculations are done using the same system of equations as in calculating the characteristic of the EES, the right-hand side of the equations having been suitably changed.

For the most common form of characteristic of the system, *at low vibration frequencies*, A_{va} is more than unity and, consequently, *the amplitude of forced vibrations decreases in cutting*. At forced vibration frequencies near to the natural frequencies ω_n of the instable forms (in which self-excited vibrations occur), A_{va} is less than unity since the corresponding points of the characteristic lie on its intersection with the negative branch of the real axis. Hence, the amplitude of forced vibrations of these resonant frequencies increases in cutting. The less the margin of stability of the system, i.e., the greater the intercept Re°_{dis}, the greater the degree of this increase in amplitude. In other words, *the resonant amplitudes of forced vibrations in cutting are always larger than when no cutting takes place.*

The *amplitude* A_{δ} *of forced vibrations, initiated due to the variability of the undeformed chip* (see Fig. 200), is equal to the geometrically specified amplitude A_{Δ} of undeformed chip thickness variation, divided by the stability factor in respect to adjustment. The required values are determined on the same gain-phase characteristic of the disconnected system. Thus

$$A_{\delta}=A_{\Delta}\frac{A_{dis}}{A_{va}} \tag{246}$$

The amplitude values (modules) of vectors A_{dis} and A_{va} are taken for the given frequency of undeformed chip thickness variation.

The ratio of the vector modules may be either less or more than unity. Accordingly, the amplitudes of vibration in cutting turn out to be greater or less than the given amplitude of variation of the machining allowance on the blank or of the undeformed chip removed by the tool (milling cutter, broach, grinding wheel, etc.).

As in the case of external force actions in machine tools with the most common form of characteristic, low frequency vibrations in cutting are reduced. The amplitude of vibrations at frequencies near to the natural frequencies of the instable forms, will vary in accordance with the stability of a system determined by the intercept Re°_{dis}.

At $Re^{\circ}_{dis}=|\,0.5\,|$, vectors A_{dis} and A_{va} will be equal to each other and therefore the amplitude of forced vibrations does not change; at $|\,1\,|>$ $>Re^{\circ}_{dis}>|\,0.5\,|$, the amplitude of the vibrations will increase. In contrast to force disturbances in a sufficiently stable system, when $Re^{\circ}_{dis}<|\,0.5\,|$, forced vibrations at natural frequencies can be substantially reduced in

cutting. In other words, a machine tool possessing high stability will operate under resonance conditions, for instance when the frequency with which cutter teeth start a cut is equal to the natural vibrational frequency of the system, as smoothly and quietly as if there were no resonance. This is known to machine tool operators who consider that they operate "good" or "bad" machine tools.

One of the important cases in practice of vibrations due to the variability of the machining allowance is the vibrations initiated upon a repeated pass along the traces (chatter marks) left by the preceding pass (see Fig. 200). If the stability is low, the vibrations initiated in the first pass will grow with each consecutive pass. The less the stability, the higher the rate of this growth. This phenomenon can be called "seesawing" the system and is one of the specific forms of loss of stability in cutting in machine tools. This case has been considered above with consideration for a delayed interlink.

The error of the machined workpiece in the presence of forced vibrations can be shown by using the concept of accuracy improvement. Two reasons for variability in the undeformed chip cross section should be differentiated: variability of the machining allowance on the blank and variability produced by the cutting tool (runout of a milling cutter, grinding wheel, etc.). In the first case, the workpiece error is determined by the displacements of the system: the larger these displacements, i.e., the amplitude of the forced vibrations, the greater the machining error will be. Accuracy improvement is determined, in this case, by the ratio

$$\frac{A_{\Delta}}{A_{\delta}} = \frac{A_{va}}{A_{dis}} \tag{247}$$

which can be transformed, for a frequency of disturbance equal to zero, quite readily into equation (244) concerning accuracy improvement for static error in one pass.

What occurs in the second case is entirely different. The error is transmitted to the workpiece from the tool and is equal to the difference between the variation in the undeformed chip due to the tool and the displacement of the system. The more the system is displaced (deflected), i.e., the larger the amplitude of the forced vibrations, the smaller the span or pitch of the waves on the workpiece surface. Practically, this means that *it is impossible to eliminate the carry-over of errors from the tool to the workpiece by increasing the processing rigidity of the system.*

A reduction in rigidity, however, is impermissible since, with a reduction in the workpiece error, metal removal is simultaneously reduced, not to mention other detrimental results of a reduction in rigidity. Hence a more radical measure is to eliminate the errors of the cutting tool.

In operation "along chatter marks", the values of the stability factors in equations (246) and (247) are taken according to the characteristic W_{dis}^{τ}

of a disconnected system with time lag, i.e., A^{τ}_{dis} is taken instead of A_{dis}, and A^{τ}_{va} instead of A_{va}.

Equation (244) is the basis for the "production" method of determining the static characteristic or the processing rigidity of the equivalent elastic system, described in Sec. 12-3. The value Δ being given and K_P being known, it is possible to measure δ and to calculate K_{EES}. Equations (245) and (247) can be employed to determine stability factors experimentally. The stability factor in respect to a variation of adjustment in the working process (cutting) is equal to accuracy improvement in periodical variation of the machining allowance. The stability factor of the system in respect to external action on the EES is determined as the ratio of the amplitude A_{ir} of machine tool vibrations in an idle run to the amplitude A_f of vibrations in cutting at the frequency ω_n of the potentially instable form. Registering these amplitudes with the aid of low-inertia apparatus, and varying the parameters of the system (cutting conditions, rigidity, etc.), the dependence of the stability factor on the various parameters can be determined. For example, by increasing the depth of cut (width of the undeformed chip in turning) we can increase K_P and reduce the degree of stability of the system. In this case, the amplitude A_δ of vibrations at the natural frequency ω_n will continue to grow until, at the maximum depth of cut ($A_{va} = 0$), it theoretically becomes infinitely large. The nonlinearity of the system limits this growth.

The above eliminates much vagueness in the interpretation of the character of machine tool vibrations, in particular those of grinding machines. The occurrence of external actions does not allow a machine tool dynamic system to be self-contained. Forced vibrations at the natural frequency of the system, observed in idle-run machine tool tests, are amplified when the degree of stability is reduced in cutting and are usually interpreted to be self-excited vibrations. This is incorrect. The level of these forced vibrations may be very high. They can be reduced by eliminating the sources of disturbance and by raising the degree of stability of the system.

Beyond the boundary of stability, we must deal with complex vibrations of a non-self-contained, self-excited oscillations system subject to external actions. It is said that there is interaction between self-excited vibrations and forced vibrations in a nonlinear system.

The *transient dynamic error* of the system determines the magnitude of the machining error in transient processes. Mentioned in the foregoing was the effect of the degree of stability and the nearness of the frequency of the forced vibrations to the natural frequency of the system on the magnitude of the dynamic machining error in the form of waves on the machined surface. This property of the system has the same effect on the size of the surface roughness in the case of other kinds of external action.

The transient dynamic error is estimated by the maximum deviation obtained in constructing the transient process. This construction is carried out in accordance with the known characteristic of the disconnected system, the given influence and the initial conditions. It is convenient to determine the transient process by simulation in an electronic analog computer.

A number of graphical methods have been developed for constructing transient processes.

A convenient procedure is the "trapezoidal method" of constructing a transient process on the basis of the real frequency characteristic of the disconnected system. This characteristic can be constructed with the aid of the so-called circle diagrams. Shown in Fig. 201 is the determination of the parameters of the real characteristic Re_y, by means of circle diagrams, for a lathe. The characteristics of the disconnected system of this lathe, in respect to cutting, are plotted on the diagrams. The characteristics have been calculated for three values of the undeformed chip width. The circle diagrams and trapezoidal methods are described in detail in many textbooks on automatic control theory and will not be given more attention here.

Figure 202 shows the real characteristic Re_y, constructed for an undeformed chip width $b = 0.6b_{lim}$, where b_{lim} is the limiting undeformed chip width. Shown also is the way the characteristic is broken down into trapezoids and the construction of the transient process for each trapezoid. The sum of these transient processes gives the required transient process that occurs when the tool suddenly begins the cut. Of interest is the fact that the transient processes due to the tool beginning the cut differ, for the various undeformed chip widths, not only by the maximum deviation, which increases with the undeformed chip width, but by the rate of decay. The damping decrement of vibrations in the system decreases with an increase in the undeformed chip width, i.e., with a reduction in the degree of stability.

Speed of response of a system. This index characterizes the speed with which a given transient process is completed. The role of quick-response in tracer- and numerical-control machine tools is well known. In these cases, it determines to a large extent the accuracy to which complex outlines and contours are machined with a specified productivity. This index is also essential for systems of automatic machining-quality control. The quick-response index, however, is no less important for general-purpose machine tools.

A specific feature of machining, distinguishing the dynamics of machine tools from the dynamics of other machinery, is the possibility of improving the accuracy (within certain limits) by consecutive repeated passes over the same part of the workpiece. This is the above-mentioned machining in several passes (cuts), the "sparking-out" process in grinding, or tool "dwell" in infeed operations of automatic and semiautomatic lathes. Machining

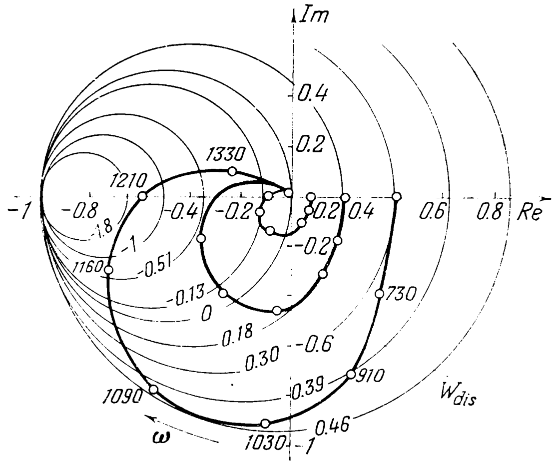


Fig. 201. Determining the parameters of the real frequency characteristic Re_y of a closed-circuit system by means of circle diagrams

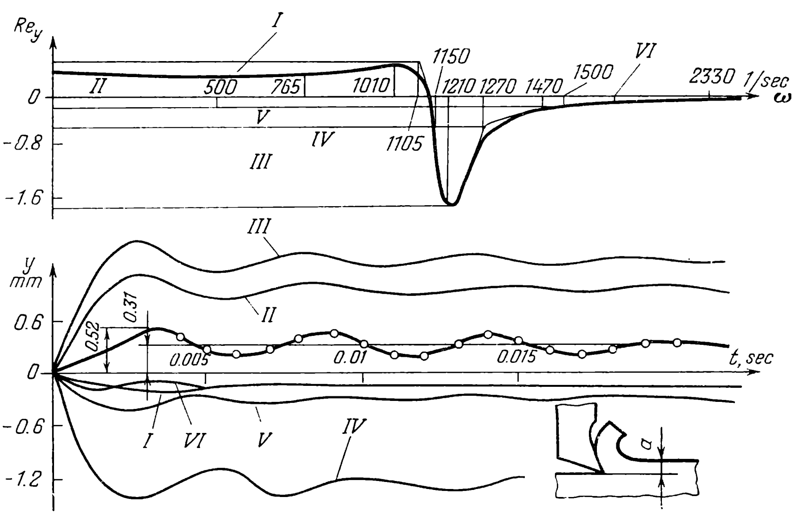


Fig. 202. The real frequency characteristic Re_y of a closed-circuit system and the transient process that occurs when the tool suddenly starts the cut

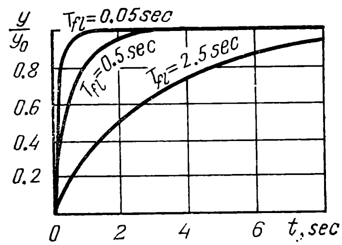


Fig. 203. The transient process in the "flotation" of a unit upon a sudden increase in sliding velocity

in one and several passes, as a stationary process, has already been considered. "Sparking-out" and "dwell" are typical representatives of transient processes due to cutting tools beginning and running out of the cut. The errors that appear when a tool starts or runs out of a cut are determined by the difference in the deflections of the elastic system at zero and full nominal undeformed chip thickness.

This error is gradually eliminated by consecutive passes of the cutting tool. At the same time the deformation of the system changes from one steady state condition to another. The duration of the transient process considerably exceeds the time for one revolution of the workpiece (one pass).

Analyzing the transient process when the tool enters or leaves the cut as a process in a system with an additional delayed interlink having a large time lag, it can be shown that it proceeds according to the exponential function

$$a = a_0 \left(1 - e^{\frac{t}{T_c}}\right) \tag{248}$$

where a = actual undeformed chip thickness
a_0 = given undeformed chip thickness

$T_c = \frac{\tau}{2}(1 + 2K_{EES}K_P)$ = accuracy improvement time constant.

The transient process can be considered to be complete at an error of 5 per cent for time $t = 3T_c$ and an error of 1 per cent for a time $t = 4.6T_c$.

Upon traversing the units of the machine tool under conditions of mixed friction, the variation in the friction forces and in the position of the units in the process of their "flotation" on the layer of lubricant may lead to the occurrence of errors in positioning the units and to machining errors. Fig. 203 illustrates the transient process for a step variation of the velocity of motion at three values of the flotation time constant T_{fl}. According to experimental data for power units of unit-built machine tools $T_{fl} \cong 0.5$ to 1 sec and the transient process may take several seconds.

In conclusion, it is necessary to note the following. The experience gained by ENIMS since 1958 in carrying out dynamic calculations for a number of different machine tools enables two stages of this work to be distinguished: preparatory work and the calculations proper.

The preparatory stage consists in drawing up the dynamics equations, resorting to possible means for their simplification. A calculator with special training and experience is required for drawing up the equations and developing the design diagram. The complexity and time-consuming nature of the preparatory stage indicate that it would be expedient to develop standard design diagrams, control systems and programmes for computers. Such a standard design diagram is to be developed for a group of machine tools having an identical layout or arrangement of their units and the same system of equations. As more experience is gained it will be possible to go over to calculations in the design of special-purpose machine tools.

The calculations proper are to be carried out for a definite design version of a machine tool, fixture or cutting tool if at least preliminary drawings are available, or of the parameters of a drive or cutting conditions.

In this stage, the initial data (masses, coefficients of the interlinks, time constants, etc.) are calculated, using the data sheets developed in the first stage and computers. On the basis of these data and a standard programme on an electronic digital computer, the frequency characteristics are calculated. Their analysis provides the values of the indices of dynamic performance of the machine tool systems. Employing an up-to-date universal digital computer, the calculations proper require from several hours to several days. This enables dynamic calculations to be used in the process of designing a new machine tool, fixture or cutting tool, as well as in selecting a drive or machining conditions (cutting speed, feed and depth of cut).

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TO THE READER

Mir Publishers would be grateful for your comments on the content, translation and design of this book. We would also be pleased to receive any other suggestions you may wish to make.

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